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Secretary..........................R. A. Lowe
Treasurer..........................R. S. Farr
Board of Governors: J. G. DeFlon, J. S. Earhart, Leo Hungerford, W. O. Stewart

Utah

Headquarters, Salt Lake City
Meets: First Wednesday

President..........................H. G. Richardson
Vice-President.....................J. T. Young, Jr.
Secretary-Treasurer................E. V. Gritton
Board of Governors: R. C. Brown, D. B. Holford, C. E. Murdock, Alfred Richeda

Washington, D. C.

Headquarters, Washington, D. C.
Meets: Second Wednesday

President..........................W. H. Littleford
Vice-President.....................L. B. Nye
Secretary..........................Lester Maurer
Treasurer..........................M. F. Hoppe
Board of Governors: H. H. Hill, J. W. Markert, F. B. Sale

Western Michigan

Headquarters, Grand Rapids
Meets: Second Monday

President..........................H. J. Metzger
Vice-President.....................J. W. Miller
Secretary..........................H. W. Wolters
Treasurer..........................K. E. Robinson
Board of Governors: H. D. Bratt, H. R. Limbacher, Frank Harbin, Jr.

Western New York

Headquarters, Buffalo
Meets: Second Monday

President..........................F. A. Moesel
1st Vice-President.................Herman Seelbach, Jr.
2nd Vice-President.................G. E. Adema
Secretary..........................J. S. Meyer
Treasurer..........................B. C. Candee
Board of Governors: M. C. Beman, Joseph Davis, Roswell Farnham, W. R. Heath, S. W. Strouse

Wisconsin

Headquarters, Milwaukee
Meets: Third Monday

President..........................O. A. Trostel
Vice-President.....................E. W. Gifford
Secretary..........................J. R. Vernon
Treasurer..........................M. W. Bishop
Board of Governors: I. J. Haus, C. W. Miller, H. W. Schreiber

1264

No. 1264

FIFTY-FIRST ANNUAL MEETING, 1945

Boston, Mass.

THE 51ST Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, held in Boston, Mass., at the Hotel Statler, was outstanding in the number of technical committee meetings that were held and the attention given to the 14 papers presented at the four technical sessions. With a registration of 611, of which 403 were members, it was evident that interest in the Society's work was widespread.

Sunday, January 21st, was largely devoted to committee meetings, final meetings of the Council and Committee on Research. Monday morning was devoted to meetings of technical committees and the Chapter Delegates meeting.

FIRST SESSION—MONDAY, JANUARY 22, 2:00 P.M.

The first session was called to order by Pres. S. H. Downs, Kalamazoo, Mich., in the Georgian Room, at 2:00 p.m. A word of welcome was given by Earl G. Carrier, General Chairman, Committee on Arrangements, in behalf of the Massachusetts Chapter.

In his opening remarks President Downs stated that the Society had suggested an international meeting for the purpose of standardizing psychrometric data and that the first meeting of this group would be held during the Annual Meeting in Boston. He read a radiogram from the *Institution of Heating and Ventilating Engineers* of Great Britain as follows:

> CONGRATULATIONS ON FORMATION OF INTER-SOCIETY COMMITTEE. BEST WISHES FOR SUCCESS OF BOSTON MEETING. REGRET REPRESENTATION IMPOSSIBLE. HAVE APPOINTED CORRESPONDENT. IHVE TABLES PRIMARILY CONFORM WITH BRITISH AIR MINISTRY METEOROLOGICAL OFFICE PUBLICATIONS. CORDIALLY
> OSCAR FABER

President Downs gave his report as follows:

PRESIDENT'S REPORT

The closing of the fiftieth year of our activity as an engineering Society brings us face to face with the greatest opportunity in our history to promote the arts and sciences of heating, ventilating and air conditioning and also the general welfare.

The responsibilities and potentialities coincidental with this opportunity have been recognized by your officers, committees, chapters and the general membership during the past year with the resulting widespread interest in what is being done and what is planned.

This interest has been clearly expressed by the increase of membership by about 25 per cent to a new high of approximately 4000, and the simultaneous increase of chapters to a total of 37. The new chapters are located in Denver; Columbus, Ohio; Memphis; Salt Lake City and Syracuse.

The Research Laboratory has been successfully put into operation in its new location at Cleveland and there has been established a forward looking, sound and detailed program which any reader of the recently published booklet will recognize as a purposeful and able approach to the needs in our field.

THE GUIDE continues to hold its great importance in the affairs of the Society and is the recognized authority in its field. It produces revenue without which the Society could not operate on its present scale.

Our finances are on a sound basis and properly liquid in accordance with constitutional requirements.

The Society and its members have continued to support the prosecution of the war by every possible means; and a great deal of our present planning will contribute largely to conversion to peacetime pursuits. A significant action in this respect is our invitation to a number of technical societies and organizations to appoint representatives to an International Joint Committee on Psychrometric Data, which has received prompt, complimentary support. A proposal for an additional joint committee on the subject of Industrial Ventilation and Air Conditioning is in the making.

The continuous publication of technical papers in the JOURNAL and the reports of chapter activities therein evidence a most satisfactory carrying forward of the objectives of the Society and the usefulness of the speakers' bureau to the chapters.

The detailed reports of other officers and the several committees present a picture of careful stewardship in their assigned duties and a planned mustering of the assets and capabilities of the Society that is essential to the realization of our hopes for the future.

In the last three years, I have been able to visit practically every chapter, including our newest ones, and I am continually impressed with the unselfish, cooperative efforts put forth by the chapter organizations and the individual members. In view of the benefit obtained by visiting chapters during the two years as vice-president, I adopted the policy this year of having the first and second vice-presidents visit a number of the chapters. This policy is beneficial to the Society as a whole, to the chapters and the individual officers, in that it creates a longer period of direct contact between the chapters and the officers, and releases the president for appearances before other organizations, which function will be of increasing importance in the future.

Our Society is still the only engineering Society having its own laboratory and such a complete program of research work. It has pioneered in this field for 25 years. The results have been good up to now, but the possibility of proper expansion of the scope and quality is definitely limited by an organizational defect which places the responsibility for obtaining research funds in the hands of a Research Finance Committee formed each year by the chairman of the Committee on Research. This causes a discontinuity of policy and action, producing very mediocre financial returns, the overall result being that practically all the research money comes from members' dues or from net balances from our publications.

Right now, while we are preparing the ground work for post war conditions, is the proper time to correct this defect in our organization. The most business-like solution appears to be to place the responsibility for research fund solicitation in the directors of the Society, namely, the Council, who would naturally combine this with other business matters handled by the secretary. This would supply the necessary link to permit formulating a continuing and long range program which is absolutely necessary if our research work is to step forward to meet larger potentialities. Further, the chairman and other Committee on Research members would be freed to concentrate on their prime problems, *i.e.*, research and the use of the funds.

In closing, I wish to express my great appreciation of the willing cooperation of the officers, the Council members and all the committee members, who have given

freely of their time and talents under these stressful, wartime conditions; and to thank A. V. Hutchinson, Secretary, Cyril Tasker, Director of Research, and others employed by the Society for duties well performed.

It is quite apparent that our Society does have before it A Greater Future than Its Great Past.

Respectfully submitted,

S. H. Downs.

The Secretary read the report of the Council:

Report of Council

During the year 1944 the Council held five meetings. At the organization meeting on February 3, at the Hotel Pennsylvania, New York, President Downs announced his appointment of Council and Special Committees. Due to a vacancy on the Council, Brig.-Gen. W. A. Danielson was selected to serve for the unexpired term of one year.

Secretary A. V. Hutchinson and Technical Secretary Carl H. Flink were reappointed; the certified public accountant was selected and the Council approved depositories for Society funds in New York, Brooklyn, and Toronto.

A membership campaign was planned, and the proposal was discussed for permanent headquarters for the Society which resulted in the appointment of a special committee to study the matter. At the newly organized Research Laboratory of the Society in Cleveland the Council met on April 30, and heard a report from President Downs regarding his visits to a group of ten chapters and to three other informal Society groups.

The program for the Semi-Annual Meeting at Grand Rapids was approved. Regulations for the method of furnishing speakers recommended by the Chapter Relations Committee were approved. A renewal lease at the New York headquarters' office was approved for a two-year period and a new five-year editorial contract with Keeney Publishing Co. was authorized.

A report on the current status of research work was made and the Council inspected the new Laboratory facilities and attended a reception sponsored by the Northern Ohio Chapter.

The New York Chapter contributed a surplus from the 50th Annual Meeting for the establishment of a fund for a permanent building for the Society and the Council created a building fund.

At Grand Rapids reports were given on the financial condition of the Society; an invitation was accepted to hold the 51st Annual Meeting in Boston and the Membership Committee was complimented on its accomplishments in increasing membership. New rules for the establishing of Society Codes and Standards were adopted by the Council.

Five nominees for a three-year term on the Committee on Research were selected.

Petitions for the organization of five chapters were acted upon during the year and new chapters were established, Rocky Mountain in Denver, Central Ohio in Columbus, Memphis, Utah in Salt Lake City and Central New York in Syracuse.

On October 16 the Council met at Columbus, and participated in the Charter Meeting of the Central Ohio Chapter.

A budget for the fiscal period November 1, 1944, to October 31, 1945, was adopted with an estimated income of $205,085.00 and an indicated expenditure of $210,105.00. Fares for Chapter Delegates and Council Members attending the Annual Meeting in Boston were authorized. In accordance with the Constitution, the Council waived the dues of men in the Service during 1945 and established a special service fee for those who desired to receive the Society's publications.

During the year at each meeting the Council acted upon resignations submitted, and cancelled the membership of those who were in arrears.

Respectfully submitted,

The Council.

Following the Reports of the President and the Council, President Downs called upon Treas. L. P. Saunders, Lockport, N. Y., for his report, which was followed by the report of the Secretary.

REPORT OF THE TREASURER

The Society is in strong liquid financial position. All investments are now in Government funds, this being accomplished during the past year. Management of the Society is more concerned with the program than finance. Appreciation is to be acknowledged for the assistance of Dr. B. M. Woods, chairman of the Finance Committee. The total assets of the Society $142,359. General Fund (ten-month period) $42,482. Reserve Fund $54,855. Endowment Fund $25,378. F. Paul Anderson Fund $1,021. Permanent Building Fund $7,351. A total matured value of bonds $123,300. Market value $99,809. Research Fund $27,381.

Respectfully submitted,

L. P. SAUNDERS, *Treasurer.*

REPORT OF THE SECRETARY

The first half century of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has just passed into history. Only two members have been privileged to participate in its activities for the full span of 50 years and one of them is here with us today, Homer Addams.

This fiftieth year has been notable in that it marks the high point of Society membership and the election of the greatest number of members in a single year. The number of chapters now is 37 with five chartered this year by the Council. Great credit is due the Membership Committee for its work, under the able leadership of M. F. Blankin, Philadelphia, Pa. The current status of membership is:

MEMBERSHIP STATUS AS OF JANUARY 1, 1945

Charter Members	2
Honorary Members	2
Life Members	93
Members	2146
Associates	1315
Juniors	272
Students	25
Total	3855

A special word of praise should be given to the Administration and Advancement Committee for its painstaking efforts in processing over 750 applications and grading the applicants.

It is difficult for any member to visualize the great amount of detail work that must be done to elect a member under our rules. After receipt of an application and until election there are 10 steps to be taken that involve writing not less than 10 letters, making an abstract of the application and sending out 40 forms and ballots, filling in 10 record cards and making over 50 entries in our books and records. In total there must be communication with at least 30 people during the election period. Although there is record of many studies by committees to simplify the procedure, there can be little accomplished without changes in the Constitution and By-Laws.

The report of the Finance Committee will give some clue to the work necessary to handle the financial affairs of the Society under the various funds set up to operate

general activities, research and special projects. Reserves were adjusted to meet the Constitutional requirements of October 31, 1944, as directed by the Finance Committee. In general, operating expenses have been held within the budget provided and income exceeded estimates.

The extent of the work of the Technical Secretary is reflected largely in the reports of the Guide Publication Committee, the Standards Committee and the Publication Committee, which are responsible for compiling the text, codes and standards of The Guide and publication of meeting and other technical papers.

With the expansion of research and other technical committee activities, the increase in membership and many wartime restrictions that have been encountered, there has been a heavier load placed on the headquarters' office staff. This was carried on without adding to the staff, although there have been numerous changes in personnel, amounting to 40 per cent.

New equipment had to be obtained for office records and the plan of establishing a uniform system for chapter records was completed this year.

There was close cooperation with the Chapter Relations Committee in handling some of the details of chapter speaker assignments and arrangements were made for the official visits of officers to all chapters.

Some of the highlights of the year's activities have been mentioned, but it would be impossible to give a complete account of all the details which must be handled at the headquarters' office.

All routine actions required by the Constitution and By-Laws have been taken with regard to membership, finances, publications and committee activities, and many special assignments have been undertaken as required by the Council.

It is a distinct pleasure for me to say to you that all of this could not have been accomplished without the full cooperation and loyalty of my associates on the staff who met the heavy demands of an unprecedented volume of work with commendable spirit and effective teamwork.

In closing one of the most successful years in the Society's history your Officers can recall with great personal satisfaction that the 50th Annual Meeting was a notable gathering which focused attention on *our great past* and pointed the way toward *a greater future.*

All who had a part in carrying on the Society's work during this 50th year appreciate the honor and I am very glad to acknowledge the helpful assistance and cooperation of the Officers, Council, Committees and Chapter Officers during this year.

Respectfully submitted,

A. V. Hutchinson, *Secretary.*

B. M. Woods, Chairman of the Finance Committee, presented his report which was followed by the Accountants' Report.

Report of Finance Committee

During the past year the Finance Committee has undertaken to appraise the financial condition of the Society and its policies of using its funds, with special reference to future development. After a statistical study we now have on curve sheets a graphical picturization of the financial history of the Society, the growth of resources year by year over the period of its life of 50 years and a picture of the growth in membership; of expenditure and income and resources each year and of cumulative resources.

Those figures have been prepared and we are undertaking to see that they are kept up to date year by year, so that it will be easy for any member of the Society to appraise its general condition.

The rapid growth in membership during the past several years prompted a study of the membership and of the average age of the membership. It is apparent to anyone who studies the histories of societies that unless the average age of the Society is kept reasonably low, the Society runs down of its own accord, and that is a matter for concern.

By means of a bar chart the number of present members of the Society who became members in any given year are shown.

Since 1934 we have recovered from the depression and have a very considerable number entering each year. In fact, during the past two years the membership has increased by roughly 950 members.

On another sheet we have had the data analyzed in the Secretary's office to show the year of birth of every member of the Society in a plotted curve. There are some 3800 points on this curve. It is slightly idealized at the moment, but the general conclusion is that the average age of the membership of the Society is somewhere around 45. I haven't had a chance yet to do the necessary averaging but it will be done. We shall try to get a few check figures—five years ago and ten years ago average age increase—to see where we stand.

The Society has under consideration by the Finance Committee and a subcommittee the question of an employee benefit plan with special reference to technical personnel in the Research Laboratory and to personnel at headquarters. The study is in progress, and I have no report to make at this time, except that the subject is a complex one and requires considerable study before any rational conclusions can be reached.

Two large questions are now before the Society that have important financial significance. One is the question of whether the funds of the Society are in sufficiently available form to be put to such uses as the Society may wish. For example, the principal funds are the Endowment Fund, the Reserve Fund and the General Fund. The limitations are as follows: Only the income of the Endowment Fund may be used by the Society and it would require a change of the Constitution to invest any portion of the Endowment Fund in anything but securities of the United States government. For example, the fund could not be used in whole or in part for a headquarters building or home for the Society even if the Society wished to use it.

With reference to the Reserve Fund, the Reserve Fund is, under the Constitution to be accumulated to the amount of $15 per member minimum, and certain Society income is to be assigned to that purpose. That means, therefore, that as the membership grows the Reserve Fund grows. The limitations upon the Reserve Fund are, first, not more than 20 per cent may be expended in any year and that only for emergencies. Then, at least half of the Reserve Fund itself must be invested in securities of the United States and Canadian Governments and the other half in investments approved for savings banks and trust companies of the State of New York. Those are wise provisions, so hedged that the Society cannot use them for any other purpose, because that would require a constitutional amendment. Experience with amendments indicates that it takes perhaps two years to make one, so that if a sudden decision is made, the Society is in a tight spot.

The questions before us in the next year or two, and they are serious questions, are: (1) Shall we make those funds more available for use—aside, of course, from the easy utility of the General and Research Funds? (2) Is a permanent and continuing policy to assure a liberal support of research, worthy of the traditions and activities of the Society? This subject has been under joint study by the Research Committee and Finance Committee and it is a pleasure to say that a special study of the administrative operation of such a plan is now being made by the Society. The Finance Committee at least hopes to have the Council consider very shortly more definite and far-reaching plans for this continuing support.

Respectfully submitted,

B. M. WOODS, *Chairman.*

Accountants' Report

TUSA & LaBELLA

CERTIFIED PUBLIC ACCOUNTANTS
52 WILLIAM ST., NEW YORK

AMERICAN SOCIETY OF HEATING AND
VENTILATING ENGINEERS,
51 Madison Avenue,
New York, N. Y.

Gentlemen:

Pursuant to your request, we examined the books of accounts and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y., and the related funds for the fiscal year ended October 31, 1944, and submit herewith our report.

The audit covered a verification of the assets and liabilities as of the close of business October 31, 1944, and a review of the operating accounts for the period then ended. For the period audited the recorded cash receipts were traced into the depositories; the cancelled bank checks were inspected, compared with the cash records and supported by payment vouchers; also the dues income and interest income from savings accounts and securities were accounted for.

A Balance Sheet reflecting the financial condition of the Society as of the close of business October 31, 1944 is submitted herewith and your attention is directed to the following comments thereon:

CASH

Cash on Deposit was verified by direct communication with the commercial and savings banks listed in the attached schedule of cash and the balances reported to us were reconciled with those shown by the books of the Society.

Checks representing the Cash on Deposit were inspected by us and the Petty Cash counted.

MARKETABLE SECURITIES

The securities, shown on the subjointed schedule, were verified by direct communication with the Bankers Trust Company, where same are deposited for safe-keeping. This asset has been included in the Balance Sheet at the cost of acquisition plus the accumulated and accrued interest earned thereon.

CERTIFICATE OF INDEBTEDNESS

The Certificate of Indebtedness issued to the Society was verified by inspection of the instrument.

The installments and interest on this indebtedness are being paid as due.

ACCOUNTS RECEIVABLE

A trial balance of the membership dues receivable was taken as of the close of business October 31, 1944. The unpaid dues were classified as to class of membership and aged as follows:

Members	$1,951.50
Associates	1,190.50
Juniors	90.00
Students	19.00
Total	$3,251.00

Dues invoiced during 1944	$2,772.00
Dues invoiced during 1943	72.00
Dues invoiced during Prior Years	407.00
Total	$3,251.00

We also took a trial balance and classified and aged the Miscellaneous Accounts Receivable which follow:

Guides	$1,374.57
Transactions	9.00
Books and Reprints	12.05
Emblems	2.50
Total	$1,398.12
Charges made during October, 1944	$ 849.49
Charges made during September, 1944	128.22
Charges made during August, 1944	103.25
Charges made prior to August 1, 1944	317.16
Total	$1,398.12

The reserves for doubtful dues and miscellaneous accounts receivable, as adjusted, in our opinion are ample to cover probable losses that might result in the collection of the same.

INVENTORIES

The emblems on hand were counted by us, and the inventories of paper and TRANSACTIONS were verified by communication with printers. These inventories were priced and computed by us. The following TRANSACTIONS were reported to us:

Volume	Year	Quantity	Price	Amount
Prior	Prior	1350	$1.00	$1,350.00
45	1939	139	1.66	230.74
46	1940	74	1.25	92.50
47	1941	43	1.32	56.76
47	1941	150 (unbound)	.96	144.00
48	1942	9	1.42	12.78
48	1942	150 (unbound)	1.07	160.50
			Total	$2,047.28

PERMANENT ASSETS

Furniture, fixtures and library are shown herein at the book values without appraisal by us. We did, however, provide for depreciation of furniture and fixtures at the rate of ten per cent per annum.

DEFERRED CHARGES

We have deferred to future operations one-sixth of all the subscriptions paid to *H.P.A.C.*, since the payment of same are on a calendar year basis and the fiscal year of the Society ends on October 31.

ACCOUNTS PAYABLE

All purchase invoices found on file that were applicable to the operations of the current fiscal period were listed and the proper liability therefor reflected in the attached Balance Sheet.

TAXES WITHHELD

The sum of $1,085.06 represents taxes withheld from employees on salaries paid during the month of October, 1944 and the additional compensation set up as of October 31, 1944.

Deferred Income

The prepaid dues, classified as to membership, follow:

Members	$ 365.65
Associates	359.50
Juniors	99.15
Total	$ 824.30

The above prepaid dues by elected members as well as the dues prepaid by candidates for membership in the additional sum of $393.13 are reflected on the Attached Balance Sheet as deferred income.

Reserve for Publications

In accordance with provisions made both in the 1943 and 1944 budgets, we have reserved the sums of $3,400.00 and $3,500.00, respectively, to cover the publication of Transactions, Volumes 49 and 50.

Funds

An analysis of the following Funds reflecting the changes that occurred in these accounts during the fiscal year ended October 31, 1944, is included herein:

General Fund
Reserve Fund
Endowment Fund
F. Paul Anderson Fund
Permanent Building Fund

The Permanent Building Fund was created by virtue of the following resolutions which were adopted at the Council meeting of October 16, 1944:

"That in accordance with Article B-xi, Section 8 of the By-Laws, that the reserve fund shall be increased to an amount equivalent to $15.00 per member as of October 31, 1944, by transfer of money in the General Fund to the Reserve Fund."

"That the amount of Initiation Fees in excess of the sum required to meet the requirements of the By-Laws for the Reserve Fund as of October 31, 1944, be allocated to the Permanent Building Fund."

The contributions to the said Permanent Building Fund during the fiscal year ended October 31, 1944, follow:

Initiation fees collected during the fiscal year ended October 31, 1944		$6,355.00
Amount of Reserve Fund in excess of $15.00 per member as of October 31, 1944		186.61
Contributions made by:		
New York Chapter	$561.06	
Western Michigan Chapter	249.27	810.33
Total		$7,351.94

There is included herein a complete financial report as prepared for the Committee on Research setting forth the financial position of the Research Fund as of the close of business October 31, 1944, and the results from operations for the fiscal year then ended.

During the current fiscal year a final liquidating dividend of $6.86 was received from the Bank of U. S. in Liquidation, leaving a balance of $145.67 which we have written off to the Endowment Fund Account.

Respectfully submitted,

Tusa & LaBella (Signed),
Certified Public Accountants.

Dated December 27, 1944.

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.
October 31, 1944

ASSETS

GENERAL FUND				
CASH				
On deposit		$21,522.73		
On hand		100.00	$21,622.73	
INVESTMENTS (AT COST)				
Securities (Market Value $23,095.00)		22,900.00		
Add: Accumulated interest	$ 1,050.00			
Add: Accrued interest	105.00	1,155.00	24,055.00	
CERTIFICATE OF INDEBTEDNESS			92.45	
ACCOUNTS RECEIVABLE				
Membership dues		3,251.00		
Less: 40% for Research	1,256.80			
Less: Reserve for doubtful	500.00	1,756.80		
		1,494.20		
Advertisers and sundry debtors	1,398.12			
Less: Reserve for doubtful	100.00	1,298.12	2,792.32	
INVENTORIES				
TRANSACTIONS—Copies	2,047.28			
TRANSACTIONS—Paper	411.11	2,458.39		
GUIDE 1945—Paper		2,217.18		
Emblems		384.25	5,059.82	
EXCHANGE			6.75	
PREPAID TRAVELING				
Railroad scrip			24.46	
PERMANENT				
Library		300.00		
Furniture and fixtures	2,923.97			
Less: Reserve for depreciation	1,742.41	1,181.56	1,481.56	
DEFERRED CHARGES				
Prepaid *H.P.A.C.* subscriptions			1,065.64	$ 56,200.73
RESERVE FUND				
Cash on deposit			3,863.30	
Securities at cost (Market value $52,548.00)		50,719.06		
Add: Accumulated interest	1,790.50			
Add: Accrued interest	33.75	1,824.25	52,543.31	56,406.61
ENDOWMENT FUND				
Cash on deposit		2,579.41		
Cash on hand for deposit		43.00	2,622.41	

Endowment Fund (*continued*)				
Securities at cost (Market value $23,166.00)		22,005.65		
Add: Accumulated interest	696.50			
Add: Accrued interest	53.84	750.34	22,755.99	25,378.40
F. Paul Anderson Fund				
Cash on deposit			8.98	
Securities at cost		1,000.00		
Add: Accrued interest		12.50	1,012.50	1,021.48
Permanent Building Fund				
Cash on hand for deposit			5,800.33	
Due from Reserve Fund			1,551.61	7,351.94
				$146,359.16

LIABILITIES AND CAPITAL

General Fund				
Liabilities				
Accounts Payable			$ 1,643.64	
Taxes Withheld			1,085.06	
Accrued Accounts				
Additional compensation to employees			3,162.34	
Deferred Income				
Prepaid membership dues	$ 824.30			
Less: 40% prepaid to Research	290.06	$ 534.24		
Dues prepaid by candidates for membership		393.13	927.37	
Reserve for Publications				
Transactions (1943) Vol. 49		$ 3,400.00		
Transactions (1944) Vol. 50		3,500.00	6,900.00	
Total Liabilities			$13,718.41	
General Fund			42,482.32	$56,200.73

Note "A": This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

Reserve Fund				
Due to Permanent Building Fund			1,551.61	
Principal		$52,819.49		
Unexpended income		2,035.51	54,855.00	56,406.61
Endowment Fund				25,378.40
F. Paul Anderson Fund				
Principal			1,008.98	
Unexpended income			12.50	1,021.48
Permanent Building Fund				
Principal				7,351.94
				$146,359.16

STATEMENT OF INCOME AND EXPENSES

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

(*For the Fiscal Year Ended October 31, 1944*)

INCOME

INCOME FROM MEMBERS				
DUES—RENEWALS				
Members and Associates	$47,556.00			
Less: Cancellations	2,792.00	$44,764.00		
40% to Research Fund		17,905.60	$26,858.40	
Juniors and Students		1,473.00		
Less: Cancellations		335.00	1,138.00	$27,996.40
DUES—NEW MEMBERS				
Members and Associates		6,925.00		
Less: 40% to Research Fund		2,770.00	4,155.00	
Juniors and Students			277.50	4,432.50
TOTAL DUES				$ 32,428.90
OTHER INCOME				
Initiation Fees			6,355.00	
Emblems and Certificate Frames			277.45	6,632.45
TOTAL INCOME FROM MEMBERS				$ 39,061.35
INCOME FROM PUBLICATIONS				
Editorial Contract			$15,999.96	
GUIDE Sales and Advertisements—Per Schedule attached			54,013.07	
Sale of TRANSACTIONS			730.34	
Income from Books, Reprints, etc.			482.81	71,226.18
INCOME FROM INVESTMENTS				
Interest—Savings Accounts			104.88	
Interest—Securities			456.17	
Interest—Certificate Frames			2.64	563.69
TOTAL INCOME				$110,851.22

EXPENSES

OPERATING EXPENSES	
President's Fund	$ 2,179.10
Council travel and meetings	2,894.84
Membership Committee	2,435.65
Admissions and Advancement Committee	246.04

Expenses (*continued*)

Constitution and By-Laws Committee	214.43		
Nominating Committee	64.72		
Chapter Relations Committee—Chapter records	400.76		
War Service Committee	66.15		
Chapter Development Committee	186.50		
Chapter Delegates' travel	2,815.24		
A. S. A. Membership	100.00		
Membership certificates	199.62		
Medals and awards	320.35	$12,123.40	
Meeting Expenses			
Meetings	3,221.30		
Speakers to Chapters	716.77		
Chapter meeting allowance	900.00	4,838.07	
Publication Expenses			
Members' subscriptions *H.P.A.C.*	6,294.53		
Guide publication and distribution—per schedule attached	48,949.28		
Transactions	3,689.47		
Membership roll	1,599.98		
Standards (including Codes)	33.99	60,567.25	
Headquarters' Expenses			
Salaries—Secretary and staff	21,995.88		
Additional compensation	3,952.93		
Traveling—Secretary and staff	1,160.76		
Rent and light	3,635.62		
Telephone	766.57		
Telegraph	199.62		
Postage	2,158.76		
General printing	971.69		
Office supplies	657.43		
Addressing and address changes	143.30		
Professional services	800.00		
Bank charges	78.98		
Depreciation of furniture and fixtures	292.40		
General office expense	892.68		
Total Headquarters' Expenses	$37,706.62		
Less: 30% to Guide	11,311.99	26,394.63	103,923.35
Net Income			6,927.87
Deduct: Council Appropriations			
Initiation fees to Permanent Building Fund		6,355.00	
Special appropriation to Research		10,000.00	16,355.00
Excess of Expenses over Income for the Year			$ 9,427.13

STATEMENT OF INCOME AND EXPENSES—GUIDE

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

(*For the Fiscal Year Ended October 31, 1944*)

INCOME			
GUIDE advertisements	$22,868.25		
Less: Discounts	518.31	$22,349.94	
GUIDE copy sales		31,663.13	
TOTAL INCOME			$54,013.07
COST OF 1944 GUIDE			
EDITING AND PRINTING			
Paper	4,065.56		
Composition and presswork	10,258.04		
Binding	4,170.00		
Mailing, expressage and cartons	666.89		
Engraving and art work	441.44		
Editorial salaries	4,116.04	23,717.97	
ADVERTISING SALES PROMOTION			
Salaries	5,556.62		
Traveling, etc.	1,670.03	7,226.65	
COPY SALES, PROMOTION AND DISTRIBUTION			
Printing, multigraphing, etc	1,895.29		
Mailing, expressage and postage	4,132.16		
Indexing	77.67	6,105.12	
GUIDE COMMITTEE EXPENSE		587.55	
TOTAL		37,637.29	
APPORTIONABLE EXPENSES		11,311.99	
TOTAL COSTS AND EXPENSES			48,949.28
NET INCOME FROM GUIDE OPERATION TO RESEARCH FUND			$ 5,063.79

BUDGET COMPARISON—SOCIETY ACTIVITIES

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

(*For the Fiscal Year Ended October 31, 1944*)

INCOME

	Actual	Budgeted	Increases	Decreases
MEMBERSHIP INCOME				
DUES—RENEWALS				
100—Members	$ 30,071.50	$ 29,930.00	$ 141.50	
101—Associates	14,692.50	17,820.00		$ 3,127.50
102—Juniors	1,100.00	3,200.00		2,100.00
103—Students	38.00	125.00		87.00
	45,902.00	51,075.00	141.50	5,314.50

Dues—New Members				
104—Members	3,639.00	1,980.00	1,759.00	
105—Associates	3,286.00	1,980.00	1,306.00	
106—Juniors	265.00	400.00		135.00
107—Students	12.50	75.00		62.50
	7,202.50	4,435.00	3,065.00	197.50
Total Dues	$ 53,104.50	$ 55,510.00	$ 3,206.50	$ 5,512.00
Other Income				
108—Initiation fees	6,355.00	2,750.00	3,605.00	
109–110—Emblems and Cert. Frames	277.45	50.00	227.45	
	6,632.45	2,800.00	3,832.45	—0—
Total Income from Members	$ 59,836.95	$ 58,310.00	$ 7,038.95	$ 5,512.00
Income from Publications				
115—Editorial contract	$ 15,999.96	$ 16,000.00		$ .04
116—Guide sales and advts.	54,013.07	46,000.00	$ 8,013.07	
117—Sale of Transactions	730.34	600.00	130.34	
118—Income from books, reprints, etc.	482.81	250.00	232.81	
119—Sale of Codes		25.00		25.00
	71,226.18	62,875.00	8,376.22	25.04
Income from Investments				
125—Interest—Savings accounts	104.88	200.00		95.12
126—Interest—Securities	456.17	750.00		293.83
127—Interest—Cert. of Indebt.	2.64	2.00	.64	
	563.69	952.00	.64	388.95
	131,626.82	122,137.00	15,415.81	
Collection of Prior Year's Dues	4,227.00	2,741.00	1,486.00	
Total Income	$135,853.82	$124,878.00	$16,901.81	$ 5,925.99

EXPENSES

Operating Expenses				
150—President's Fund	$ 2,179.10	$ 2,500.00		$ 320.90
151—Council travel to meetings	2,894.84	2,500.00	$ 394.84	
160—Executive Committee		100.00		100.00
161—Finance Committee		200.00		200.00
162—Membership Committee	2,435.65	2,500.00		64.35
170—Admissions and Adv. Comm.	246.04	150.00	96.04	
171—Constitution and By-Laws Comm.	214.43	150.00	64.43	
172—Nominating Comm.	64.72	150.00		85.28
173C—Chap. Rel. Comm.—Chap. records	400.76		400.76	
174—War Service Comm.	66.15	1,500.00		1,433.85

EXPENSES (*continued*)

175—Chapter Development Comm.	186.50		186.50	
173B—Chapter Delegates travel	2,815.24	2,300.00	515.24	
201—A.S.A. membership	100.00	100.00		
204—Membership certificates	199.62	200.00		.38
206—Medal and awards	320.35	200.00	120.35	
	12,123.40	12,550.00	1,778.16	2,204.76
MEETING EXPENSES				
163—Meetings	3,221.30	3,000.00	221.30	
173A—Speakers to Chapters	716.77	1,200.00		483.23
327—Chapter meeting allowance	900.00	900.00		
	4,838.07	5,100.00	221.30	483.23
PUBLICATION EXPENSES				
200—Members subscription *H.P.A.C.*	6,294.53	6,000.00	294.53	
305-326—GUIDE publ. and distrib.	48,949.28	44,060.00	4,889.28	
202—TRANSACTIONS	3,689.47	3,500.00	189.47	
203—Membership roll	1,599.98	1,500.00	99.98	
164—Standards (including Codes)	33.99	150.00		116.01
	60,567.25	55,210.00	5,473.26	116.01
HEADQUARTERS' EXPENSES				
210—Salaries—Secy. and staff	21,995.88	21,500.00	495.88	
211—Additional compensation	3,952.93	3,000.00	952.93	
212—Traveling—Secy. and staff	1,160.76	1,200.00		39.24
213—Rent and light	3,635.62	3,600.00	35.62	
214—Telephone	766.57	700.00	66.57	
215—Telegraph	199.62	200.00		.38
216—Postage	2,158.76	1,600.00	558.76	
217—General printing	971.69	900.00	71.69	
218—Office supplies	657.43	600.00	57.43	
219—Addressing and address changes	143.30	200.00		56.70
220—Professional services	800.00	800.00		
221—Bank charges	78.98	50.00	28.98	
222—Depreciation—Furn. and Fixt.	292.40	250.00	42.40	
223—General office exp.	892.68	600.00	292.68	
TOTAL HEADQUARTERS' EXPENSES	37,706.62	35,200.00	2,602.94	96.32
Less: 30% charge to GUIDE	11,311.99	10,560.00	751.99	
	26,394.63	24,640.00	1,850.95	96.32
INITIATION FEES TO BUILDING FUND	6,355.00	2,750.00	3,605.00	
COUNCIL APPROPRIATION TO RESEARCH	10,000.00	10,000.00		
TOTAL EXPENSES	$120,278.35	$110,250.00	$12,928.67	$ 2,900.32

CONDENSED FINANCIAL REPORT

Research Fund

(Oct. 31, 1944)

Assets	
Cash on hand and on deposit	$27,650.02
Laboratory equipment, furniture and fixtures	6,359.53
Prepaid rent	1,325.00
Total Research Fund	$35,334.55
Endowment Fund	588.06
	$35,922.61
Liabilities and Funds	
Accounts payable and taxes	$ 1,417.86
Unexpended balance of Council appropriation	3,401.95
Deferred income—earmarked projects	2,842.70
Prepaid dues for 1945	290.06
Research Fund	27,381.98
Total Research Liabilities and Fund	$35,334.55
Endowment Fund	588.06
	$35,922.61

ANALYSIS OF RESEARCH FUND

Balance November 1, 1943			$28,749.54
Additions			
Refund—University of Wisconsin	$ 1,185.85		
Equipment, furniture and fixtures	6,601.27	$ 7,787.12	
Deductions			
Excess of expenses over income for fiscal year ended Oct. 31, 1944		9,154.68	
Net Reduction			1,367.56
Balance Oct. 31, 1944 per Balance Sheet			$27,381.98

STATEMENT OF INCOME AND EXPENSES

(For the Fiscal Year Ended October 31, 1944)

INCOME

From A.S.H.V.E.		
40% of 1944 and prior year's dues collected	$21,503.60	
Special Council appropriation	6,598.05	$28,101.65
Contributions—Per Schedule		
General	$ 4,803.00	
Earmarked	316.75	
Glass Study—balance from prior year	1,088.38	6,208.13

INCOME (*continued*)

INTEREST FROM BANK DEPOSITS			37.63
TOTAL INCOME			$34,347.41

EXPENSES

CHAIRMAN AND COMMITTEE			
Traveling	$ 673.07		
Chairman's office	102.31		
Professional services	200.00	975.38	
RESEARCH LABORATORY			
Salaries—administrative and technical	$16,101.83		
Salaries—clerical (N. Y. and laboratory)	3,901.33		
Traveling	1,385.87		
Postage	283.82		
Telephone and telegraph	308.56		
Office expenses and supplies	746.13		
Printing and mimeographing	950.00		
Library and periodicals	217.30		
Laboratory materials and supplies	1,504.61		
Insurance	231.01		
Depreciation—equipment and fixtures	340.71		
Unallocated	133.10		
Taxes (Cleveland rent)	1,855.00		
Property insurance	288.55		
Heating	385.00		
Electricity	86.50		
Gas	12.96		
Water	13.50		
Janitor	516.40		
Building maintenance	16.48	29,278.66	
COOPERATIVE RESEARCH			
University of California	$ 1,600.00		
University of Minnesota	1,200.00		
Case School of Applied Science	2,000.00		
Oregon State College	500.00		
Texas A & M College	250.00		
Cornell University	500.00		
University of Pennsylvania	600.00	6,650.00	
SPECIAL COUNCIL APPROPRIATION			
Expended part of $10,000.00 Grant		6,598.05	43,502.09
EXCESS OF EXPENSES OVER INCOME			$ 9,154.68

The next report was that of the Membership Committee by the Chairman. M. F. Blankin, Philadelphia, Pa.

REPORT OF MEMBERSHIP COMMITTEE

In January, 1944, President Downs did a very unusual thing. He asked a Past President to do something and wanted me to take the chairmanship of the Membership Committee. I accepted because I was vitally interested in increasing the membership of the Society.

When I came into office in 1943 I found that we had been on a steady downgrade in membership. We had reached a peak in 1940 of about 3,200 and as of January,

1943, we were down to 3,006 members. I felt that it was not difficult to attract members to our Society; we had so much to offer.

As President, I went around the country preaching the Society and what we had to offer and, as a result, the membership was increased during that year to an all-time high.

Last January when I suggested that the potential membership of the Society might be 5,000, some people thought that was grossly exaggerated. It looks to your committee as if it were slightly pessimistic.

During the past year we have had an exceptional growth. We received applications from 751 persons for membership in the Society. Between January 1st and December 31st we elected 752 new members. We have at the present time on hand a total of approximately 125 applications for membership that have not been processed. As of January 19th the total membership of the Society is 3,938, a very substantial increase of about 33⅓ per cent in two years.

I might state that the chairman and the Membership Committee do not deserve the credit for this increase. The chapters and the members of the Society have really made a substantial contribution to that increase in membership and they are the ones who are to be commended for it.

During the past year five new chapters were admitted into the Society: Central New York, Central Ohio, Denver, Memphis and Utah. As a result of those five chapters we secured approximately 250 new members, so you can see that the older chapters and the members contributed a great share of the new applications. It can continue this year. There is an opportunity for several new chapters this year and I ask that you give your hearty support to the new Membership Committee and see if you cannot gradually approach the 5,000 ideal. It will mean a lot to the Society. We shall be able to do much more for all members and still more in the way of research, which finally is the thing in which we are all interested.

Respectfully submitted,
M. F. Blankin, *Chairman.*

President Downs suggested that during 1945 the Society could easily reach a membership of 5000.

L. F. Collins, Detroit, Mich., presented his paper on Studies of the Mechanism of Solution of CO_2 in Condensates Formed in the Steam Heating Systems of Buildings (see p. 39) and D. S. McKinney, Pittsburgh, Pa., presented the paper Preventing the Solution of CO_2 in Condensates by Venting of the Vapor Space of Steam Heating Equipment, prepared by D. S. McKinney, J. J. McGovern, C. W. Young and L. F. Collins (see p. 53).

President Downs thanked the authors and those who discussed the paper for the excellent presentation and comments and then introduced Prof. E. B. Watson, Ithaca, N. Y., who presented an abstract of the papers Summer Weather Data and Sol-Air Temperature—Study of Data for New York City, by C. O. Mackey and E. B. Watson (see p. 75) and Summer Weather Data and Sol-Air Temperature—Study of Data for Lincoln, Neb., by C. O. Mackey (see p. 93).

President Downs announced that A. O. May, Chicago, Ill., would present a resolution from the Chapter Delegates Committee.

Mr. May presented the following resolution, passed by the Chapter Delegates Committee, for the purpose of improving the method of selecting Society nominees for office:

The Nominating Committee shall consist of eleven members of the Society qualified to vote, who shall have held membership in the Society for at least the five previous years. Seven members of the committee and an alternate to be elected by the Chapter Delegates in attendance at the meeting of the Chapter Delegates Committee to be

held the first day of the Annual Meeting. Four members of the Nominating Committee and an alternate to be elected by the Council at the previous Fall Council meeting.

It shall be provided that no member of the Nominating Committee shall be a member of Council, and that no two members of the committee shall come from the same Chapter area, and at least one from each regional area to be established.

Round trip railroad fare and lower berth shall be paid to the members of the Nominating Committee in attendance at the Semi-Annual Meeting where the final selection of nominees shall be made.

It was moved by Mr. Jones and seconded by Mr. Blankin:

> **THAT the resolution be referred to the Council for transmission to the Constitution and By-Laws Committee with instructions to present the resolution in proper form for Society action at a later meeting.**

Mr. Avery reminded the members present that by previous action at the Annual Meeting in Cincinnati the Society had voted against a Constitutional Amendment which would reduce the Nominating Committee from 31 to 6 members. He advocated that the motion be voted down unless the members intended to vote for the resolution when the Constitution and By-Laws Committee would present it for Society action.

The motion was approved by a 52 to 38 vote.

President Downs called for the Report of the Tellers of Election for Officers and Committee on Research and C. S. Koehler, New York, N. Y., presented the report as follows:

BALLOTS FOR OFFICERS

President—C.-E. A. Winslow	1161
1st Vice-President—Alfred J. Offner	1163
2nd Vice-President—W. A. Russell	1160
Treasurer—L. P. Saunders	1165
Members of Council (*three-year term*)	
W. A. Danielson	1161
H. R. Roth	1165
Ernest Szekely	1162
B. M. Woods	1162
(Scattering votes for other candidates)	
Committee on Research (*three-year term*)	
R. M. Conner	1165
John A. Goff	1165
F. W. Hutchinson	1164
W. E. Zieber	1162
(Scattering votes for other candidates)	
Total Ballots Received	1242
Total Legal Ballots	1165

The meeting adjourned at 5:45 p.m.

SECOND SESSION—TUESDAY, JANUARY 23, 9:30 A.M.

The second session convened at 9:30 a.m. with Vice-Pres. C.-E. A. Winslow, New Haven, presiding.

Chairman Winslow called for the Report of the Committee on Research which was presented by Prof. G. L. Tuve, Cleveland, Ohio, *chairman*.

Committee on Research Annual Report, 1944

The efforts of the Research Committee in 1944 were centered upon the technical planning of a program that could be rapidly expanded at the end of the war. There was substantial progress on actual research during the year, but adequate personnel were not yet available either at the Cleveland Laboratory or at the cooperating institutions. The research booklet, published in October, gave the Society membership a progress report, and also furnished a basis on which industry could be invited to participate in our enlarged program.

Research work has been in progress during 1944 on at least twenty of the projects outlined in the booklet, and a number of new projects are in the planning stage. Additional cooperative research is under consideration with several universities. Special attention has been paid to the correlation and coordination of our program with similar research elsewhere, and support has been given to the formation of inter-society joint committees in the fields of Psychrometry and of Industrial Ventilation.

During 1943, the Committee on Research, under the chairmanship of C. M. Ashley, formulated an eight-point program for increasing the scope and magnitude of research operations. Three of these, the securing of a Director of Research, amendment of research regulations and securing a new research home, were accomplished in 1943, and in March, 1944, Director Tasker moved the Laboratory to its new location in Cleveland. Three more of the eight objectives were substantially realized in 1944, *viz.*, development of a long-range research program, enlargement of cooperative research with the colleges, and encouragement of participation by industry. The two remaining steps in the program, the expansion of Laboratory staff and the coordination with other technical activities of the Society, were carried as far as circumstances would permit. The research staff now consists of eight full-time members including the Director.

During the Cleveland gas fire disaster, our Laboratory found an opportunity to render unique service. The *American Gas Association* Testing Laboratories were immediately adjacent to the liquid gas tanks where the fire originated. Hence, the Testing Laboratory staff escaped only with great difficulty and their losses were most serious. The space and facilities of a portion of our building were at once offered and accepted, as a base from which their work of rehabilitation could be directed.

An eminent contribution to the success of our entire research program has been made by Samuel R. Lewis and the Research Finance Committee. Not only was the total of industry participation increased over that of the last several years, but 49 firms increased their subscription over those of the previous year and 17 other firms participated who had not before had a part in the Society's research program.

The major credit for all features of the 1944 Research Program is due the Society's Director of Research, Cyril Tasker. The Annual Report of the Director to the Committee on Research is transmitted as a part of this report.

The response of both members and friends of the Society to the challenge of the enlarged research program has been most gratifying. Many assisted in the preparation of the 1944 Research Booklet. Many others, recipients of the 4000 odd copies of the booklet, have made additional suggestions for technical research. For this encouragement, the chairman and all members of the Committee on Research are most grateful.

Statistics

Meetings	
Committee on Research	3
Research Executive Committee	6
Research Finance Committee	2
Technical Advisory Committees	27
Chapter Research Presentations	7
Research Projects and Papers	
Projects under way at Cleveland Laboratory	8
Projects under way at Cooperating Institutions	13
Proposals received for new cooperative projects	7
Research reports and papers received	10

SUMMARY FINANCIAL REPORT

November 1, 1943 to October 31, 1944

INCOME			
From member dues and bank interest		$21,541	
Council appropriation		6,598	
Industry contributions:			
General (1944)	$5,120		
Earmarked (1944) for future studies	983		
Carried forward from 1943 Earmarked Funds	1,088		
		7,191	
			$35,330
EXPENSES			
Payments to Cooperating Institutions (7)		6,650	
Expenditures at Pittsburgh, at Cleveland Laboratory, and at New York office as per detailed financial report. (See Report of Director of Research)		36,852	
			$43,502

INDUSTRY PARTICIPATION

October 1, 1944 to January 15, 1945

4 Contributions of $1,000.00 or more	$ 4,500
10 Contributions of 500.00 to $1,000.00	5,000
15 Contributions of 200.00 to 500.00	3,400
37 Contributions of 100.00 to 200.00	3,818
21 Contributions below $100.00	790
87	$17.508

Earmarked,	$ 6,950.00
General	10,558.00

In the year 1944 we have advanced some distance on the road towards the complete reorganization and reorientation of Society research.

Viewed in the light of Society history, the high lights of the year will probably be shown as (*a*) the removal of the Research Laboratory from the Bureau of Mines Building in Pittsburgh to its new quarters in Cleveland and (*b*) the preparation and distribution of a research booklet setting forth the details of the first group of projects in the Society's expanded research program. We believe that in spite of obstacles sound research work has been carried on at the Laboratory as well as at many of the cooperating institutions and that the work accomplished will stand in the record after we have forgotten the removal.

For convenience the first part of this report will be divided into a number of main headings dealing with work under the various Technical Advisory Committees of the Society, commencing with a report on their organization and selection.

TECHNICAL ADVISORY COMMITTEES

During 1944 there were in existence 21 Technical Advisory Committees. At the 50th Annual Meeting of the Society held in New York in January, 17 of these committees met to assist in formulating the research program.

Early in the year a careful survey was made of the membership of these committees in order to bring into active participation in Society research men who had shown marked interest and ability for such work. In general an endeavor was made to form committees with a membership of about 10, but some committees covered such wide interests that a larger membership was found desirable.

The Research Laboratory now handles the duplication and distribution of all minutes of Technical Advisory Committee meetings and a duplicate ballot form has been printed to assist in the more speedy handling of papers and reports submitted to the committees for comment and approval. An attempt has also been made to distribute from time to time to the membership of the committee's technical material which has come into the possession of the Laboratory and which, it was felt, might be of interest in the committee's activities.

A new committee was formed in the spring to deal with *Air Sterilization and Odor Control* and we were fortunate to obtain the consent of Dr. W. F. Wells, University of Pennsylvania Medical School, to act as chairman.

Air Cleaning *—R. S. Dill, *Chairman*

The committee met at the 50th Annual Meeting in New York and again at the Semi-Annual Meeting in Grand Rapids. At the winter meeting a sub-committee was appointed to investigate and report on methods of testing grease filters. The sub-committee submitted its report at the summer meeting and the Technical Advisory Committee then recommended that this report be studied by the Committee on Research with a view to arranging for the carrying out of additional tests preparatory to the preparation of a code for the testing of grease filters.

The committee was advised in January that the Air Filter Institute had appointed an Engineering Committee to develop standards and codes for specific air filter applications. A progress report was submitted to the committee in June and the tentative suggested code is being submitted to the Technical Advisory Committee at the Boston Meeting (January, 1945). One meeting of this committee of the Air Filter Institute was held in September, on the invitation of the Committee on Research, at the Cleveland Laboratory.

The studies under Professor Rowley at the University of Minnesota have continued during 1944, though at a reduced rate due to staff shortages and the priority demands for research for the Armed Services.

Some work has been done on the analysis of different types of filters to obtain the pressure drop through these filters as they become filled with dust and further work has been carried out on the photometric method of rating and testing air filters.

Air Conditioning in Industry *—W. L. Fleisher, *Chairman*

The committee met in January at New York and recommended to the Committee on Research that laws, codes, standards and recommendations in heating and ventilating in industrial establishments (not including laws on toxic vapors) be collected and summarized at the Research Laboratory for this committee to analyze.

Soon after the removal of the Laboratory to Cleveland, a letter was sent out to the various state and municipal authorities requesting them to furnish the Society with copies of all appropriate codes and regulations. The response was very encouraging and to date 167 different codes and standards have been assembled. It is not to be expected that all of these codes contain data pertinent to the study being made, but all will have to be reviewed since in many cases useful information is found where it is least expected.

So far forty-six codes have been reviewed; of these, forty contain definite regulations concerning heating, ventilating or air conditioning and abstracts of these regulations have been made. This study will be accelerated as staff becomes available.

Air Distribution and Air Friction *—Prof. D. W. Nelson, *Chairman*

Research work under this committee has had to give way to other work in most of the institutions at which it was being carried out. That it is a very important subject, having wide Society and industrial interest, was shown by the many comments received at the Laboratory on the three projects dealing with air flow which were listed in the Research Booklet.

A meeting of the committee was held at Grand Rapids and it was recommended that, because of criticisms received by the Society regarding the present Guide chart for Friction of Air in Pipes, a critical review of all available information, published or otherwise, on this subject be reviewed by a competent engineer working either at the Research Laboratory or elsewhere under the direction of the Committee on Research.

The committee also agreed on the imperative need for more data on the resistance to the flow of air encountered in the wide variety of duct fittings now in use throughout the industry. The Society has been approached by other organizations interested

* Complete committee personnel published in Front Section of 1944 Transactions.

in this subject with a view to coordinating investigations in the fields covered by this committee.

Work is now in progress at Case School of Applied Science, under a cooperative agreement with the Society, on air distribution from diffusing grilles, slots and perforated plates.

It has not been possible to date to engage suitable staff to work on these problems at the Laboratory; it is hoped that this condition may soon be remedied for it is apparent that there is a great deal that could be done to obtain more satisfactory data for use by the membership of the Society.

AIR STERILIZATION AND ODOR CONTROL *—Dr. W. F. Wells, *Chairman*

This committee, formed in the spring of 1944, held an informal organization meeting at Grand Rapids and will meet again at Boston. There is considerable interest in the work of this committee as evidenced by the papers presented at the Summer Meeting and by the many references in the literature to means for air sterilization and the prevention of the spread of respiratory diseases. The modern tendency to reduce uncontrolled infiltration to a minimum has produced problems in odor control and our research work must assist in finding satisfactory solutions of this problem.

COOLING LOAD IN SUMMER AIR CONDITIONING *—W. E. Zieber, *Chairman*

The committee met both at the 50th Annual Meeting and at the Semi-Annual Meeting. Two papers from cooperative research at Cornell University have been presented to the Society under its sponsorship:

1. Periodic Heat Flow through Homogeneous Walls or Roofs, by C. O. Mackey and L. T. Wright, Jr. (Published in TRANSACTIONS, Vol. 50, 1944, p. 293.)
2. Summer Air Conditioning—Summer Weather Data for Lincoln, Nebraska, by C. O. Mackey (published in this issue, p. 93).

Another paper, closely allied with this committee's work, entitled, Summer Weather Data for New York City, by C. O. Mackey and E. B. Watsons is also being presented at the 51st Annual Meeting in Boston.

Since studies involving mathematical analysis of heat flow through homogeneous walls and roofs could be considered as complete, at least for the time being, the committee turned its attention to the questions involved in heat transfer through nonhomogeneous walls and roofs and the mathematical analysis involved therein. Preliminary calculations have indicated that the position of the insulation (or other material with a low heat transfer coefficient) would have a material effect upon the time lag and maximum heat flow. At the same time it is desirous of arranging for some practical checks to be made on the calculations, or of obtaining sufficient verification of the factors employed to have the two methods correlated.

The committee agreed that a wall would have to be built and tested in the sun in order to determine the cycle of *sol-air* temperature and the outside heat transfer coefficients with varying wind velocities over the wall. The data so obtained can then be used in hot-box type tests to check other calculated values.

The committee is actively interested in projects 15, 16 and 17 in the Research Booklet, and is making arrangements for cooperative research work at other universities and for continuing the work at Cornell.

COOLING TOWERS AND SPRAY PONDS *—H. B. Nottage, *Chairman*

Cooperative research work at the University of California has proceeded under the direction of Professor Boelter and his colleagues. It is expected that a Technical Advisory Committee will be appointed during 1945 to deal with evaporative condensers and spray ponds and at least one other cooperative project may be arranged to study another aspect of the problem of cooling towers. The committee has been promised specific assistance to make other studies possible. It was recommended that future Society research include (*a*) more extensive year-'round weather data and (*b*) the question of standardization of instruments and methods of testing.

Corrosion*—Leo F. Collins, *Chairman*

The Research Laboratory, at the request of the committee, engaged the services of the Engineering Societies' Library to make a thorough search of technical literature and compile a bibliography from which treatises could be prepared by six members of the committee on the subjects:

1. Protective coatings which may be used to minimize corrosion of air conditioning equipment, including ducts handling moist air.
2. The relation between *bad* water and corrosion in coolers, condensers and evaporative condensers.
3. The relationship between internal corrosion of refrigerating systems and those extraneous materials which commonly contaminate refrigerants.
4. The causes for and the prevention of corrosion in cooling towers, spray humidifiers (and dehumidifiers), cooling coils and brine spray systems.
5. The cause and prevention of external corrosion in gas and oil fired heaters and connecting flues.
6. The internal corrosion of steam heating equipment.

Previously this Technical Advisory Committee had confined its attention mostly to the subject of corrosion in steam heating equipment. Two papers sponsored by the committee have been completed: Preventing the Solution of CO_2 in Condensates by Venting of the Vapor Space of Steam Heating Equipment, by D. S. McKinney, J. J. McGovern, C. W. Young and L. F. Collins and Studies of the Mechanism of Solution of CO_2 in Condensates Formed in Steam Heating Systems of Buildings, by L. F. Collins.

Flow of Fluids in Pipes and Fittings*—F. E. Giesecke, *Chairman*

A cooperative research project is in progress at the Agricultural and Mechanical College of Texas to study (*a*) the effect of pipe size on the simultaneous flow of two currents of water of different temperatures in opposite direction in a vertical pipe and (*b*) the turbulence produced in the water flowing in a pipe by a jet of a smaller stream projected into the pipe.

Fuels*—R. A. Sherman, *Chairman*

Since returns from a committee ballot had revealed a predominant desire for a fundamental and comprehensive investigation of factors covering: (1) The performance of chimneys; (2) the performance of barometric dampers; (3) the methods for measuring flue gas and surface temperatures; the Laboratory undertook a study of the performance of barometric dampers. Eight different dampers were received from manufacturers. Preliminary tests were made with the dampers connected to a horizontal duct system in which the draft was supplied by means of a fan. The dampers were then installed in the flue pipe connecting a gas fired furnace with a chimney built entirely within a Laboratory Building, so that the effect of the barometric damper could be studied when hot gases were flowing through the flue pipe. A progress report is being made to the committee at the Boston Meeting.

At the recommendation of this committee a paper, A Survey of Testing Methods and Rating Limits for Domestic Heating Devices, was prepared by R. S. Dill, and is being presented at the 51st Annual Meeting (see p. 185.)

Glass*—R. A. Miller, *Chairman*

The committee requested that the Research Laboratory "prepare a bibliography of data on heat transmission through windows and through glass-block panels, abstract these data, tabulate the results given and present the abstracts and tabulations to the committee together with comments on methods and reliability of results. The bibliography collected should include all available data on film coefficients for single and multiple glazing and heat loss or gain from shaded glass areas. Data on solar radiation relating to the above topics should be included. Data on summer and winter requirements should be segregated."

Using a literature search made by the Engineering Societies' Library as a basis, a comprehensive report has been prepared by G. V. Parmelee for review and criticism, so that the material can be published as a special research bulletin of the Committee

on Research. A summary report has also been prepared and distributed to the committee membership. Suggestions have been made as to future research work which should be carried out to clarify the points on which present data appear to be inadequate or lacking.

HEAT REQUIREMENTS OF BUILDINGS *—P. D. Close, *Chairman*

The committee is on record *as being opposed in general to the use of exposure factors as such* and recommends *that the problem be considered under the heading of Infiltration, since wind generally has only a relatively minor effect on the transmission losses, except in the case of glass transmission losses or heat losses through thin, highly conductive materials.*

The committee agreed that there was an urgent need for more accurate infiltration data on actual buildings under operating conditions. At a meeting held in Grand Rapids in June, the committee agreed on three proposed methods of measuring infiltration as follows: (1) A plenum chamber pressure difference method using simple equipment suggested by C. M. Ashley; (2) By measuring the rate of decay of concentration of CO_2 released into a room; (3) By measuring the total heat input and subtracting the calculated transmission loss.

Studies are presently under way in Pittsburgh at the residence of Prof. T. F. Rockwell and at the I=B=R Research Home in Urbana, Illinois, in cooperation with the University of Illinois. Funds earmarked for this study were provided by the *Insulation Board Institute* during 1944.

HEAT TRANSFER OF FINNED TUBES *—William Goodman, *Chairman*

The committee has undertaken a study of refrigerant-side film coefficients and their effect upon the overall performance of direct expansion air coolers. A cooperative experimental program is being carried on at Case School of Applied Science. A test unit has been set up which allows the refrigerant coefficients in various parts of an evaporator to be measured. Effects of load and of the oil content of the refrigerant are being studied, as are also the effects of flash gas, of quality and of superheat.

The committee reviewed and accepted for presentation at the 1945 Annual Meeting a paper by L. G. Seigel on Air Cooling Coil Problems and Their Solutions. This paper is the result of a research program undertaken at the instigation of the Navy Department, Bureau of Ships, Air Conditioning Section and carried on at Case School of Applied Science.

INSTRUMENTS *—C. M. Ashley, *Chairman*

The Technical Advisory Committee on Instruments met at New York in January in a combined meeting with the Committee on Radiation and Comfort. Most of the discussion at this meeting centered around the measurement of radiation and comfort effects since it was felt that the development of a suitable instrument or instruments capable of measuring mean radiant temperature, or some other factor which can be correlated with comfort effect in radiant heating, was of the highest importance in future studies. The committee was very much interested in comments made by Prof. Edy Velander of Sweden, who told of similar work being undertaken in Swedish research organizations.

In order to summarize the present position, a sub-committee was appointed to prepare a bibliography on methods and instruments available for measuring radiation temperatures under comfort conditions. The sub-committee, which consisted of Messrs. Broderick, Humphreys, and Prof. F. W. Hutchinson, with Professor Fahnestock as chairman, prepared a bibliography which was later duplicated and distributed to members of the committee. It contained 26 references and was entitled, The Measurement of Effective Temperature and Mean Radiant Temperature in Rooms for Human Occupancy. Copies will be made available to any other interested parties on request.

The chairman has prepared a comprehensive list covering instruments in practically all the fields that may be of interest to members of the Society. It is the intention of this committee to analyze the instrument question very thoroughly and

set up a program of experimental work on those instruments on which it is felt work is needed in the immediate future. This comprehensive list has been distributed to all members of the committee.

Insulation *—E. R. Queer, *Chairman*

The activities of this committee were limited during 1944 to a revision of Chapter 4 of The Guide—Heat Transmission Coefficients. The test work to determine new thermal conductivity values for The Guide, which are much in demand, has been held in abeyance until a later date when more facilities can be made available.

Physiological Reactions *—Dr. R. W. Keeton, *Chairman*

The committee's chief purpose is to present, at various intervals, summaries of current physiological researches with special relation to their bearing on engineering practice.

As soon as the Armed Services are in a position to release, for the benefit of the engineering profession, the results of the many physiological studies they have made during the past three or four years, this committee intends to interpret the data for the benefit of the Society membership.

Psychrometry *—J. H. Walker, *Chairman*

The results of the cooperative research work at the University of Pennsylvania were summarized in the paper, Thermodynamic Properties of Moist Air, by John A. Goff and S. Gratch (see p. 125), for use in a proposed revision of Table 6, Chapter 1 of The Guide.

The ultimate objective of this cooperative investigation has been a formulation of the thermodynamic properties of moist air which, by reason of accuracy and thermodynamic consistency, can claim universal acceptance as standard.

At the January meeting, the committee considered a suggested program of work in hydrometry submitted by Messrs. Harrison and Little of the U. S. Weather Bureau and the committee went on record as recognizing that, following completion of current thermodynamic studies, attention should be directed to the development of suitable instruments for measuring moisture content in air, such instruments to have an accuracy comparable with the new thermodynamic data, as the committee also felt that attention should be directed to the development of a suitable technique for calibrating existing and new instruments or apparatus for measuring moisture content.

The Laboratory prepared a report on The Accuracy of Psychrometric Instruments and Their Application, to assist the committee in planning a program of experimental work. Copies of this report may be obtained from the Laboratory on request.

Inter-Society Committee

The committee, at its meeting in June, recommended to the Council that invitations be sent to all organizations likely to be interested in the problem of the properties of mixtures of air and water vapor to appoint a representative on a joint Inter-Society Committee to consider the adoption, when available, of basic equations and tables of such properties.

The Council approved this recommendation and invitations were, therefore, sent out in the name of the Society by President Downs to the following organizations:

United States:

American Society of Mechanical Engineers; American Society of Refrigerating Engineers; American Institute of Chemical Engineers; American Physical Society; Institute of Aeronautical Sciences, Inc.; Department of Commerce (National Bureau of Standards, Weather Bureau).

Canada:

Meteorological Div., Canadian Dept. of Transport; National Research Council of Canada.

Great Britain:

Institution of Heating and Ventilating Engineers; Physical Society of London; National Physical Laboratory.

The inaugural meeting was held at Boston on January 23, 1945.

RADIATION AND COMFORT *—J. C. Fitts, *Chairman*

The committee in a joint session with the Technical Advisory Committee on Instruments concluded that methods of measuring radiation and comfort effects were fundamental to all studies of radiant heating and cooling and that no instruments presently available can be considered as wholly satisfactory for use in laboratory or field studies.

As a result of cooperative research at the University of California sponsored by this committee, a paper entitled, Radiation Corrections for Basic Constants used in the Design of all Types of Heating Systems, was prepared by B. F. Raber and F. W. Hutchinson (see p. 213).

The comments received at the Laboratory have emphasized the great interest being taken in panel and radiant heating by many members of the Society and by certain sections of the industry.

SENSATIONS OF COMFORT *—Thomas Chester, *Chairman*

The restudy and analysis of data taken in 1937 during a cooperative research project between the Minneapolis-Honeywell Regulator Co. and the Society has been continued at the University of Minnesota as a cooperative project. They report as follows:

"Data from the 25,000 cards which were originally taken have been tabulated on special tabulating machine cards. The sorting and analyzing of these data is now in progress and the work is about 50 per cent complete.

"The first part of the analysis consists of a study of the effects of dry-bulb temperature and relative humidity on the sensation of comfort. From data obtained in the investigations, combinations of dry-bulb temperature and relative humidity giving equal effective temperatures are collected and curves have been plotted showing sensations of comfort as ordinates and dry-bulb temperatures as abscissae. The variations in the sensation of comfort as recorded by the test subjects is now being analyzed to find the cause of these variations. Some interesting trends are apparent. However, in many cases, even though there were 25,000 cards, sufficient fundamental data are not available to draw definite conclusions. It has been difficult for the University to maintain an adequate research staff on this program, but it is expected that the work will be completed within the next two or three months. The final analyses of the test data have involved a considerable amount of study and numerous graphs are being drawn in an effort to get the true significance of the records which were taken."

From the preliminary results it seems quite likely that further research work will be recommended with concentration on some of the more vital factors involved.

The committee met jointly with the Technical Advisory Committee on Physiological Reactions and recommended that, as soon as conditions permitted, studies be undertaken on the effect of low dew points on comfort and health and on the physiological and psychological effects of drafts.

SORBENTS *—Lt. Comdr. John Everetts, Jr., *Chairman*

The committee, formed in 1943, met at the time of the 50th Annual Meeting in New York and adopted the following program for immediate action: (1) A review of THE GUIDE chapter on cooling, dehumidification and dehydration; (2) A determination of the types of industries, processes and applications to which sorbents may be practically and economically applied; (3) A determination of interaction constants for other pressures than atmosphere; (4) Correlation of chemical and physical characteristics of commercially available sorbents; (5) Compilation of data on storage requirements of materials of all kinds which are affected by changes of humidity.

Under Item (1), revised definitions of *Dehumidification* and *Dehydration* in accordance with present practice and usage of these terms have been submitted for consideration by the Guide Publication Committee.

Under Item (4), considerable data have been accumulated during the past year on the following Sorbents: *a.* Silica Gel, *b.* Activated Alumina, *c.* Dow Chemical Co. 8A Solution, *d.* Dow Chemical Co. 17A Solution, *e.* Lithium Chloride, *f.* Calcium Chloride, *g,* Glycols and Glycol derivatives.

Correlation of these data will be made as soon as more complete information becomes available.

Preliminary work has been commenced on the other three items listed.

Sound Control *—R. D. Madison, *Chairman*

The committee has agreed to proceed with studies dealing with: (*a*) Apparatus measurement, ultimately leading to sound test codes; (*b*) Attenuation of ducts, elbows and other fittings; (*c*) Room conditions and recommended sound levels.

The committee is dissatisfied with the relatively broad tolerances with regard to sound level meters permitted under present standards, since these tolerances cause great difficulties in the practical use of such meters.

Some preliminary tests were made at the Research Laboratory on fans fitted with inlet and outlet ducts, measurements being made of the total sound generated and its effect on the room sound level. Difficulties with the Laboratory's sound level equipment caused these tests to be postponed pending a check and recalibration by an outside agency.

The cooperation of the Insulation Board Institute in generously providing a supply of rigid insulation for the construction of a test room is gratefully acknowledged by the Technical Advisory Committee and the Committee on Research.

Weather Design Conditions *—Capt. T. H. Urdahl, *Chairman*

The committee met at New York at the time of the 50th Annual Meeting and agreed that it was illogical to try to determine design temperatures based simply on extreme temperatures on record or on means of extremes for any particular month.

Since extensive data for 110 stations throughout the country could be made available to the Society through the U. S. Weather Bureau, the Committee on Research approved the recommendation of the Technical Advisory Committee that arrangements be made to have this data analyzed either at the Research Laboratory or elsewhere if more suitable arrangements could be made.

To govern the method of analysis, the following recommendations were made:

1. The design dry-bulb temperature for winter shall be taken as that temperature which is equalled or exceeded for 97½ per cent of the hours during the months of December, January, February and March. The winter wet-bulb temperature and dew-point shall be taken as those temperatures which are equalled or exceeded for 95 per cent of the hours during the same months.
2. The summer design dry-bulb temperature shall be taken as that temperature which is equalled or exceeded by no more than 2½ per cent of the hours during June, July, August and September. The design wet-bulb and dew-point temperatures for these same months shall be those temperatures which are not exceeded for more than 5 per cent of the time.
3. Wind data shall be prepared in tabulated form.

During the spring and early summer, one member of the committee, J. C. Albright, personally condensed about half of the data for one station so as to devise satisfactory speed-up techniques and methods of presenting the data. At a later date, the suggestion was made that the collected data be analyzed at the Statistical Laboratory of Iowa State College. Negotiations are now in progress with a view to setting up a cooperative research project for this purpose. The final method of presentation of the data will be decided by the Technical Advisory Committee, since it is most essential that the data be presented in a form readily usable by air conditioning engineers.

Oregon Chapter Research Advisory Committee—J. E. Yates, *Chairman*

The cooperative project on a study of *the heat flow through wet building walls* being carried out at Oregon State College is proceeding. One of the most serious difficulties during the tests has been the warping of the test wall sections when subjected to conditions simulating heavy rain. Recent improvements in the test equipment have allowed some tests to be run on the walls and the results check published data for dry still air and 15 mph wind satisfactorily.

Other Laboratory Activities

In addition to the work conducted at the Laboratory under the direction of the various Technical Advisory Committees, other Laboratory activities are discussed below.

Staff

The Laboratory was fortunate in obtaining the temporary services of Clark M. Humphreys, who acted as Senior Engineer from May 15 to October 6, when he

returned to other duties at Pittsburgh. His assistance in the difficult first months at the new Laboratory is gratefully acknowledged.

At present the Laboratory staff is made up as follows: G. V. Parmelee (on leave of absence from Fenn College, Cleveland), Research Fellow; R. G. Huebscher, Research Assistant; Doris M. Dietz, Laboratory Assistant; William L. Ryan, Mechanic; Ilse M. Jahn, Office Manager and Secretary; Lelia D. Henderson and Thelma T. Branske, Stenographers.

During the past few months we have continuously been on the lookout for additional trained technical staff. This is the most pressing problem facing the Laboratory today and one in which the assistance of all members of the committee and friends of the Society is requested.

The research booklet, published in October and distributed, first, to some 400 executives of the industry in connection with the financial appeal and, later in the month, to the entire membership of the Society, has attracted considerable attention not only in the Society but outside of it also. We have had a large number of requests for booklets and though 4250 copies were printed, it is now out of print and only a few file and reference copies remain.

It entailed a great deal of effort by the members of the Research Executive Committee and the chairmen of the various Technical Advisory Committees as well as by the Laboratory staff. The assistance given by the Secretary of the Society, A. V. Hutchinson, merits especial mention—much of the attractiveness of the make-up of the booklet resulted from his suggestions.

We have received, as a result of this booklet, a number of helpful letters suggesting other research projects of importance to the profession and industry. We believe that the booklet was of great assistance in the financial campaign in that it set out a planned research program for industry inspection and study.

The expenses incurred in the printing and distribution of the booklet were charged jointly against the General Research Fund and the special Council appropriation.

Navy Reports

Late in the summer, the Research Laboratory was advised that the Bureau of Ships was anxious to obtain several additional copies of the reports made to the Navy Department by the Society under contracts numbered 66853 and NYs 10817. After discussion with Captain Stacey and Commander Urdahl, it was decided that the reports should be retyped in such a way that they could be reproduced by photolithography.

They were, at the same time, carefully edited and rearranged so that composite reports could be prepared; this entailed considerable work by the Laboratory staff, but resulted in well arranged and concise reports. The Navy Department set up a new contract to cover the printing of these reports. The release of the information given therein, for the benefit of the Society membership, has been agreed upon by the Navy Department.

The reports will be bound in 4 books as follows:

BOOK I: Cyclic Temperature Changes in a Ship Structure Due to Sun Radiation; The Humidification of Living Quarters on Naval Ships; Air Volume Requirements for Occupants of Space on Naval Ships; Cooling Requirements of Aviator Ready Rooms on Aircraft Carriers; The Spot Cooling of Workers in Engine and Fire Rooms of Naval Ships.

BOOK II: Noise Measurement and the Physiological and Psychological Effects of Noise on Man.

BOOK III: Effect of Branch Take-Off Design on Noise in Ventilating Duct Systems; A Psychological and Physiological Study of the Accuracy, Variability, and Volume of Work of Young Men in Hot Spaces with Different Noise Levels.

BOOK IV: Physiological Response of Subjects Exposed to High Effective Temperatures and Elevated Mean Radiant Temperatures.

Cooperative Agreements

In view of the plan to enlarge the cooperative research activities of the Society, the Laboratory was instructed to make a study of all past and existing cooperative

contracts with a view to recommending a skeleton contract that might be generally applicable for future agreements.

The results of the study were transmitted to the sub-committee, consisting of Prof. L. E. Seeley, chairman; W. E. Heibel and Dr. C.-E. A. Winslow, appointed at the June meeting of the Committee on Research *to investigate the form of the cooperative contract to be drawn up between the Society and institutions with which we desire to carry out cooperative research work.* This committee will report at the final meeting of the 1944 Committee on Research.

Equipment

Office and Laboratory equipment for which a definite use could be foreseen were removed to Cleveland and most of this equipment has since been overhauled, repaired and put back into service. New equipment has been purchased or fabricated as the need arose.

Design of Psychrometric Rooms

In connection with the design of new psychrometric rooms, Mr. Humphreys visited during the summer the following laboratories having psychrometric rooms and discussed methods of construction and operation with the operating personnel:

Aero Medical Research Laboratory, Wright Field, Dayton, O.; Armoured Medical Research Laboratory, Fort Knox, Ky.; Naval Medical Research Institute, Bethesda, Md.; U. S. Institute of Public Health, Bethesda, Md.; Naval Air Experiment Station, U. S. Navy Yard, Philadelphia, Pa.; Pierce Laboratory, New Haven, Conn.; Climatic Research Laboratory, Lawrence, Mass.; Harvard Fatigue Laboratory, Boston, Mass.

Plans have been prepared showing the projected location of the proposed test rooms; some detailed drawings have been made of suggested methods of construction.

Library

At the June meeting of the Committee on Research, approval was given for the establishment at the Laboratory of an adequate library to be named *The John R. Allen Memorial Library.* A suitable bookplate is now being prepared and a general invitation will be sent to authors of text books in the field of heating, ventilating and air conditioning to donate a copy of their book or books to this library. A general invitation will be made to all members of the Society to contribute books, pamphlets and reprints to this library. All donations will be shown on the bookplate and reported in the Journal.

INSTITUTIONS COOPERATING WITH THE COMMITTEE ON RESEARCH

Agricultural and Mechanical College of Texas: Gravity Circulation of Water in a Vertical Pipe; Turbulence Due to Circulating Pumps. *Case School of Applied Science:* Air Distribution in Rooms; Heat Transfer Coefficients of *Freon* Refrigerants. *Cornell University:* Methods of Computing Solar Heat Loads of Walls and Roofs; Comparative Analysis of Psychrometric Systems. *Oregon State College:* Heat Transfer Through Wetted Walls. *University of California:* Air and Surface Temperatures in a Panel Heated Room; The Measurement of the Radiant Effect in Heated and Cooled Rooms; Cooling Tower Design and Performance. *University of Illinois (College of Engineering):* Heat Losses from Basement Walls and Ceilings; Heat Losses Due to Infiltration. *University of Minnesota:* Air Cleaning Devices; Comfort and Environment—Statistical Analysis. *University of Pennsylvania:* The Properties of Air-Water Vapor Mixtures.

RESEARCH PAPERS—1944

1. Some Effects of Attic Fan Operation on Comfort, by W. A. Hinton and W. G. Wanamaker (Atlanta) (Research Report No. 1261, A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 371).
2. Periodic Heat Flow—Homogeneous Walls or Roofs, by C. O. Mackey and L. T. Wright (Ithaca) (Research Report No. 1255, A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 293).
3. Radiation Corrections for Basic Constants Used in the Design of All Types of Heating Systems, by B. F. Raber and F. W. Hutchinson (California) (see p. 213).
4. Preventing the Solution of CO_2 in Condensates by Venting of the Vapor Space of Steam Heating Equipment, by D. S. McKinney, J. J. McGovern, C. W. Young and L. F. Collins (Pittsburgh) (see p. 53).

5. Thermodynamic Properties of Moist Air, by John A. Goff and S. Gratch (Pennsylvania) (see p. 125).

6. Summer Weather Data and Sol-Air Temperature—Study of Data for Lincoln, Nebraska, by C. O. Mackey (Ithaca) (see p. 93).

7. Air Cooling Coil Problems and Their Solutions, by L. G. Seigel (Case School) (see p. 165).

CONTRIBUTORS TO RESEARCH—1944 PROGRAM

Financial Contributions

Aerofin Corp., *Airtemp Division of Chrysler Corp., Alco Valve Co., Inc., Allegheny County Steam Heating Co., Aluminum Company of America, American Blower Corp., The American Rolling Mill Co., *Anemostat Corporation of America, Barber-Colman Co., Barnes & Jones, Inc., Bayley Blower Co., Bethlehem Steel Co., Brunner Manufacturing Co., Buffalo Forge Co., Burnham Boiler Corp., *Byers Co., A. M., Carrier Corp., *Chamberlin Metal Weatherstrip Co., Chase Brass & Copper Co., Chicago Pump Co., *Conkey, H. D., & Co., Field Control Division, Crane Co., *The Detroit Edison Co., Dole Valve Co., Dravo Corp., Driscoll, W. H., Duquesne Light Co.

Forslund Pump & Machinery Co., Frick Co., Inc., Fulton Sylphon Co., G & O Manufacturing Co., Hays Corp., *Heating, Piping & Air Conditioning Contractors National Association*, Hoffman Specialty Co., Inc., Ilg Electric Ventilating Co., Illinois Chapter of A.S.H.V.E., Illinois Engineering Co., Illinois Testing Laboratories, Inc., *Insulation Board Institute, Johns-Manville, *Johnson Service Co., Keeney Publishing Co., Kewanee Boiler Corp., Kinetic Chemicals, Inc., *La-Del Conveyor & Mfg. Co., The R. C. Mahon Co., *May Oil Burner Corp., McDonnell & Miller, McQuay, Inc., Mellish & Murray Co., Minneapolis-Honeywell Regulator Co., *Modine Manufacturing Co., Mueller Brass Co.

*Narowetz Heating & Ventilating Co., The Nash Engineering Co., The National Radiator Co., National Tube Co., *Nesbitt, John J., Inc., *Owens-Corning Fiberglas Corp., *The Perfex Corp., Pipe Fabrication Institute, Pittsburgh Corning Corp., *Pittsburgh Plate Glass Co., *Portland Cement Association*, Powers Regulator Co., The Pyle-National Co., Raisler Corp., Randall Graphite Products Corp., *The F. C. Russell Co.

*Sarcotherm Controls, Inc., Servel, Inc., The Sisalkraft Co., Spencer Thermostat Co., Sturtevant, B. F., Co., Surface Combustion, Taylor Instrument Cos., Tempil Corp., *Thrush, H. A. & Co., Timken Detroit Axle Co., *Trade Wind Motorfans, Inc., The Trane Co., *U. S. Machine Corp., Weil-McLain Co., Williams Oil-O-Matic Heating Corp., Wing, L. J., Mfg. Co., Wood Conversion Co., *Worthington Pump & Machinery Corp., York Corp., Young Radiator Co.

Contributors of Equipment, etc.

American Blower Corp., The Detroit Edison Co., Insulation Board Institute, Minneapolis-Honeywell Regulator Co., Modine Manufacturing Co., The W. R. Rhoton Co.

Chairman Winslow expressed his appreciation of the excellent work done by Chairman Tuve and Cyril Tasker, Director of Research, in organizing the research program and establishing the new Research Laboratory in effective operation.

The following message from R. M. Conner, Director, *A.G.A.* Testing Laboratories, Cleveland, Ohio, was read:

Will you please express for me, following presentation of your report tomorrow, the *American Gas Association's* thanks for the help given us especially by your Laboratory following recent disaster to our laboratories. As you know we intend to contribute to your 1945 Research Fund. Our Association also sends greetings and best wishes for the Society's Annual Meeting.

The chairman then presented W. E. Crowell, Buffalo, N. Y., who presented an abstract of his paper, Altitude Chamber for Study of Heating and Air Conditioning Problems (see p. 111).

The author pointed out that the present high plane speeds, by causing an air temperature rise due to kinetic energy given up to the surface, created a cooling problem.

Chairman Winslow suggested that Mr. Crowell's remarks might lead to a cooperative research project between A.S.H.V.E. Committee on Research and

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* Earmarked contributions.

the aircraft industry and that in such a project the Society would be delighted to be of service.

Chairman Winslow then called for the paper, Thermodynamic Properties of Moist Air, which was presented in abstract by John A. Goff, Philadelphia, Pa., (see p. 125).

Chairman Winslow joined the other commentators in paying tribute to Dean Goff's contribution to the Society. He also stated that no other society had contributed so much as had A.S.H.V.E. to the science with which it deals.

The final paper, Air Cooling Coil Problems and Their Solutions, by L. G. Seigel, Cleveland, Ohio, was presented in abstract by the author (see p. 165).

The session was adjourned at 12:30 p.m.

Third Session—Tuesday, January 23, 2:15 p.m.

The third technical session convened at 2:15 p.m., January 23, with Vice-Pres. A. J. Offner, New York, N. Y., presiding.

Mr. Offner presented J. F. Collins, Jr., Pittsburgh, Pa., Chairman of the Guide Publication Committee, who gave a report for the committee on The 1945 Guide.

Report of Guide Publication Committee

During the past busy war year, 50 authors have contributed much time and thought to writing, revising and improving the 23rd Edition of The Guide and many other members have offered valuable suggestions to the committee on its work. To determine policy, the committee held several meetings during the year, and the individual members of the committee each have acted in the capacity of sponsors for the changes being made in groups of chapters.

I will not take your time now to tell you of the work that has been done on the individual chapters. Such a list will be found in the preface to the book. However, I draw your attention to the new chapter on the important and popular subject of Panel Heating and Radiant Heating written in part by our President-Elect, Dr. Winslow, and in part by our Past President, Dr. F. E. Giesecke. This chapter has been examined by a number of members who are engaged in this field and has been pronounced as very excellent and one which will bring much credit to the Society.

You will find that while the page size has not been altered, each page in the text section of the new Guide has three more lines than in former editions. By this change, therefore, the total amount of technical text was actually increased.

It is suggested that the next Guide Publication Committee attempt to prepare chapters in a final form as nearly as possible in order to relieve the Technical Secretary of the tremendous load which develops at the end of the year if chapters are delayed and if they need much editorial revision to prepare them for the printer.

The Guide is an important activity of the Society from a financial as well as a technical viewpoint.

Respectfully submitted,
J. F. Collins, Jr., *Chairman*

Mr. Dill presented an abstract of his paper, A Survey of Testing Methods and Rating Limits for Domestic Heating Devices (see p. 185).

Chairman Offner thanked the author and the commentators, and then announced that the registration at the meeting at 2:20 p.m. was 555.

R. J. Lorenzi presented an abstract of the paper, The Influence of Heat Capacity of Walls on Interior Thermal Conditions and Heat Economy, by C.-E. A. Winslow, L. P. Herrington, and himself, New Haven, Conn. (see p. 197).

The final paper for the session, Radiation Corrections for Basic Constants Used in the Design of All Types of Heating Systems, by B. F. Raber and F. W. Hutchinson, Berkeley, Calif., was presented in abstract by Professor Hutchinson (see p. 213).

Chairman Offner thanked the authors and commentators. The meeting was adjourned at 3:30 p.m.

FOURTH SESSION—WEDNESDAY, JANUARY 24, 9:45 A.M.

The last session convened at 9:45 a.m. with Pres. S. H. Downs presiding.

President Downs introduced W. N. Witheridge, Detroit, Mich., who presented an abstract of his paper, Control of Industrial Atmospheres (see p. 227).

President Downs announced that due to the unavoidable absence of the author he would present the paper, Mine Ventilation and Its Relation to Health and Safety, by D. Harrington (see p. 243).

President Downs announced that the total registration for the Annual Meeting was 611, 403 being members.

C. S. Leopold, Philadelphia, Pa., presented an abstract of his paper, Tobacco Smoke Control—A Preliminary Study (see p. 255).

Chairman Winslow called for the presentation of the paper, Noise Rating of Ventilating Fans, by W. H. Hoppmann, II, and Fred Lager, New York, N. Y. Mr. Hoppmann presented an abstract of the paper (see p. 271).

INSTALLATION OF OFFICERS

President Downs then announced that the next order of business was the installation of the officers elected for 1945 and asked Past President W. T. Jones to preside at the Installation. He was assisted by C. V. Haynes and Homer Addams. The new President, Dr. C.-E. A. Winslow was installed.

Other incoming Society officers were installed as follows: 1st Vice-President A. J. Offner, by Past President W. H. Driscoll; 2nd Vice-President W. A. Russell, by Past President M. F. Blankin; Treasurer L. P. Saunders, by Past President W. H. Driscoll.

Past President W. L. Fleisher then installed the following newly elected members of the Council: Brig. Gen. W. A. Danielson, Memphis, Tenn.; H. R. Roth, Toronto, Can.; Ernest Szekely, Milwaukee, Wis.; and Dr. B. M. Woods, Berkeley, Calif.

Mr. Jones then declared the officers properly installed for the year 1945.

President Winslow referred to the breadth of interest shown in the subjects presented at the session and stated that they had ranged through the fields of vision in physics, optics and acoustics.

T. D. Stafford, Grand Rapids, Mich., presented the report of the Committee on Resolutions, whereupon it was moved by L. P. Saunders, seconded by L. T. Avery, and

VOTED: THAT the following resolutions be adopted:

Resolutions

WHEREAS, our President, Sewell H. Downs, has, in the first year of the second half century of our Society's life, set a fine example of leadership in both technical guidance and business management, and

WHEREAS, through his wise direction the Society's membership has been increased both in number and in unity of effort, and

Whereas, the Society's high standards so widely established by his predecessors have been upheld and promoted by his steadfast adherence to them,

Be It Resolved, the A.S.H.V.E. extends its grateful thanks and deep appreciation to him and also to Mrs. Downs, for her inspirational loyalty.

* * * *

Whereas, the members of the Council, Advisory Council, Committee on Research, Technical Advisory Committees and the Special Committees of A.S.H.V.E. have not only carried on the work of the Society within their jurisdiction, but have been the instrumentalities through which the Society's knowledge and experience have been utilized in the prosecution of the war, and

Whereas, the time, effort and knowledge of individual members of these groups have been wholeheartedly given,

Be It Resolved, that the sincere thanks of the Society be given to them.

* * * *

Whereas, some have made the supreme sacrifice,

Be It Resolved, that the A.S.H.V.E. extend greetings to those who still serve and deep sympathy to the families and friends of those who will not return, and

Be It Further Resolved, that a copy of this resolution be sent to the next of kin of those who have given their lives and who will continue with us in memory.

* * * *

Whereas, the Past Presidents of this Society have wisely and hopefully laid the foundation stone on which we now build, and

Whereas, some have found it impossible to be with us at this meeting,

Be It Resolved, that the members of the A.S.H.V.E. make known to all of them our joy in having those with us who could attend and our disappointment in not being able to greet those who were absent.

* * * *

Whereas, the finest traditions of New England hospitality have been upheld by the Committee on Arrangements of the Massachusetts Chapter at the 51st Annual Meeting of the A.S.H.V.E., and

Whereas, many individuals and organizations have contributed to the splendid success of the meeting,

Be It Resolved That an expression of thanks and appreciation be adopted and copies of this resolution be sent to the following:

To Earl G. Carrier, General Chairman of Committee on Arrangements, and to his several Committee Chairmen, and their committee members, and ladies.

To authors for the interesting technical papers.

To various Engineering Societies that sent representatives to establish the joint Inter-Society Committee on Psychrometry.

To Past Pres. William T. Jones, the Banquet Toastmaster.

To Prof. James Holt, Toastmaster of the Get-Together Dinner.

To the various members who assisted in furnishing registration personnel.

To the B. F. Sturtevant Co. for their courtesy in making the inspection trip possible.

To the organizations responsible for the Sunday afternoon broadcasts.

To the management and staff of the Hotel Statler for their courteous and excellent service.

To the newspapers and trade papers for their coverage of our 51st Annual Meeting.

Respectfully submitted,

Resolutions Committee
T. D. Stafford, *Chairman*
D. M. Allen
G. C. Kerr

As there was no further business, the 51st Annual Meeting was declared adjourned at 1:15 p.m. by Dr. Winslow.

PAST PRESIDENTS' AND GET-TOGETHER DINNER

The Past Presidents' Dinner, which was attended by Homer Addams, M. F. Blankin, W. H. Carrier, W. H. Driscoll, W. L. Fleisher, E. Holt Gurney, C. V. Haynes, W. T. Jones and J. F. McIntire, was combined with a Get-Together Dinner for members and guests on Monday evening. An authentic New England boiled dinner was served.

BANQUET

The annual banquet was held on Tuesday evening, January 23.

W. T. Jones acted as toastmaster and introduced each member seated at the head table. Short speeches were made by retiring President Downs, President-Elect Winslow and Mr. Addams, who spoke briefly on the progress of the Society during the past 50 years.

The speaker at the banquet was Charles Winthrop Copp, a Vermonter and graduate of Oberlin College, who had been a teacher of English in Japanese government schools for two decades and who was imprisoned as a spy at the outbreak of the war. Mr. Copp was released by the Japanese in June, 1944 and returned to the United States on the exchange ship Gripsholm. The speaker's subject was Japan—Yesterday, Today and Tomorrow.

PROGRAM 51ST ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
HOTEL STATLER, BOSTON, MASS.
JANUARY 22-24, 1945

Sunday, January 21

MEETINGS

11:00 A.M. Finance Committee—Parlor F.
1:30 P.M. Council Meeting—Parlor B.
4:30 P.M. Committee on Research—Parlor C.
7:00 P.M. TAC Radiation and Comfort—Hancock Room

SPECIAL EVENTS

(For Members and Ladies)

2:00 P.M. Visit to Mrs. Jack Gardner's Palace

TRANSPORTATION: Street Car or taxi (taxi fare approximately $1 per cab)

2:30 P.M. Radio Broadcast—Musical Program

LOCATION: New England Mutual Insurance Co. Hall
TRANSPORTATION: See Group Leader
NOTE: Tickets free. The supply is limited and must be obtained at 2:00 P.M. on Mezzanine
TICKETS: By application only.

4:00 P.M. Radio Broadcast—Symphony Concert—Conductor A. Fiedler

LOCATION: Boston Opera House
TRANSPORTATION: See Group Leader
NOTE: Tickets free. The supply is limited and must be obtained at 2:00 P.M. on Mezzanine
TICKETS: By application only.

Monday, January 22

9:00 A.M. Registration—Mezzanine Floor

COMMITTEE MEETINGS

9:30 A.M. Chapter Delegates Conference—Parlor C
9:30 A.M. TAC Air Distribution and Air Friction—Parlor B
TAC Glass—Parlor F
TAC Fuels
TAC Air Cleaning
TAC Air Sterilization and Odor Controls
10:00 A.M. TAC on Corrosion
10:00 A.M. Committee on Heavy Duty Furnaces
12:15 P.M. Council and Authors Luncheon
12:30 P.M. TAC Cooling Load in Summer Air Conditioning

SPECIAL EVENTS

10:00 A.M. Covered Wagon Sightseeing Trip
COST: $2.00 per person (30c. admission to Paul Revere House)

10:00 A.M. Observation Tower New Court House
TRANSPORTATION: Taxi or street car (taxi fare approximately 75c.)

10:00 A.M. Ladies Unconducted Shopping Tours
1:30 P.M. Visit to Agassiz Museum—Glass Flowers
TRANSPORTATION: Street car to Harvard Square, Taxi or street car from Harvard Square (taxi fare approximately 50c. per cab)

2:00 P.M. Technical Session—Pres. S. H. Downs, Presiding—Georgian Room
Reports of Officers and Council
Studies of the Mechanism of Solution of CO_2 in Condensates Formed in the Steam Heating System of Buildings, by L. F. Collins
Preventing the Solution of CO_2 in Condensates by Venting of the Vapor Space of Steam Heating Equipment, by D. S. McKinney, J. J. McGovern, C. W. Young and L. F. Collins
Summer Weather Data and Sol-Air Temperature—Study of Data for New York City, by C. O. Mackey and E. B. Watson
Summer Weather Data and Sol-Air Temperature—Study of Data for Lincoln, Nebraska, by C. O. Mackey
Report of Tellers of Election—C. S. Koehler, *Chairman*
6:30 P.M. Past Presidents Dinner
6:30 P.M. Get-together Supper—typical New England Dinner
7:00 P.M. Organization Meeting. 1945 Committee on Research—Parlor D

Tuesday, January 23

9:30 A.M. Technical Session—Vice-Pres. C.-E. A. Winslow, Presiding—Georgian Room
Report of Committee on Research, by G. L. Tuve, *Chairman*
Altitude Chamber for Study of Heating and Air Conditioning Problems, by W. E. Crowell
Thermodynamic Properties of Moist Air, by John A. Goff and S. Gratch
Air Cooling Coil Problems and Their Solutions, by L. G. Seigel
12:30 P.M. Nominating Committee Meeting—Parlor A
1:30 P.M. Joint Inter-Society Committee on Psychrometry—Parlor D

SPECIAL EVENTS

10:00 A.M. Same as Monday
1:30 P.M. Same as Monday
2:00 P.M. Ladies Bridge Tea

2:00 P.M. TECHNICAL SESSION—Vice-Pres. A. J. Offner, Presiding—Georgian Room
Report of Guide Committee, by J. F. Collins, *Chairman*
A Survey of Testing Methods and Rating Limits for Domestic Heating Devices, by R. S. Dill
The Influence of Heat Capacity of Walls in Interior Thermal Conditions and Heat Economy, by C.-E. A. Winslow, L. P. Herrington and R. J. Lorenzi
Radiation Corrections for Basic Constants Used in the Design of All Types of Heating Systems, by B. F. Raber and F. W. Hutchinson

4:00 P.M. TAC Instruments

7:00 P.M. ANNUAL BANQUET
Toastmaster—W. T. Jones
Address: Japan—Yesterday, Today and Tomorrow, by Charles Winthrop Copp

Wednesday, January 24

9:30 A.M. TECHNICAL SESSION—Pres. S. H. Downs, Presiding—Georgian Room
Control of Industrial Atmospheres, by W. N. Witheridge
Mine Ventilation and Its Relation to Health and Safety, by D. Harrington
Tobacco Smoke Control—A Preliminary Study, by C. S. Leopold
Noise Ratings of Ventilating Fans, by W. H. Hoppmann, II, and Fred Lager
Unfinished Business (Reports of Committees)
Installation of Officers
New Business—Resolutions
Adjournment

1:30 P.M. Inspection Trips.

1:30 P.M. Organization Meeting of Council—Hancock Room

COMMITTEE ON ARRANGEMENTS

EARL G. CARRIER, *General Chairman*
Vice-Chairmen
JAMES HOLT D. M. ARCHER

Advisory—W. T. JONES, *Chairman;* D. S. BOYDEN, F. D. B. INGALLS, F. J. TUTTLE

Banquet—G. B. GERRISH, *Chairman;* C. T. FLINT, T. F. MCCOY, W. A. MCPHERSON

Finance—J. W. BRINTON, *Chairman;* E. A. DUSOSSOIT, E. W. MCMULLEN, N. J. H. SHAW.

Transportation—F. R. ELLIS, *Chairman;* W. J. AHEARN, C. W. LARSON

Entertainment—C. M. F. PETERSON, *Chairman;* R. L. LINCOLN, C. R. SWANEY, H. L. VON REHBERG, C. P. YAGLOU, B. J. WAHLIN

Publicity—A. C. BARTLETT, *Chairman;* E. L. BLAIR, C. W. KIMBALL, H. B. WIEGNER

Ladies—A. L. HESSELSCHWERDT, JR., *Chairman;* T. P. MANDELL, L. B. MANN, H. E. PARKER

Reception and Registration—P. A. CRONEY, *Chairman;* L. A. BRISSETTE, ADOLPH EHRENZELLER, J. M. MEANS, H. E. WHITTEN

Hostesses—MRS. JAMES HOLT, *Chairman;* MMES. D. M. ARCHER, J. W. BRINTON, E. G. CARRIER, A. L. HESSELSCHWERDT, JR., W. T. JONES

1265

No. 1265

STUDIES OF THE MECHANISM OF SOLUTION OF CO_2 IN CONDENSATES FORMED IN STEAM HEATING SYSTEMS OF BUILDINGS †

By LEO F. COLLINS,* DETROIT, MICH.

WHILE corrosion is always the resultant of simple chemical reactions, most of the industrial problems so created are complicated because they embody the simultaneous operation of a number of elementary processes. To understand what takes place, one must be able to accurately appraise not only the chemical and physical factors involved but also their relative bearing upon one another as conditions change with different operating procedures. At times, all of these stipulations cannot be met but there exists, nevertheless, the necessity for *explaining* corrosion phenomena. Under such circumstances postulates must be used.

In the field of steam heating system corrosion the mechanism of the solution of carbon dioxide has been postulated for the most part. The assumptions that necessarily had to be made dealt with the physico-chemical factors presumed to control the CO_2 content of condensates. It was noted frequently, however, that the CO_2 content of condensates, drained from operating systems, were far greater than theory predicted and it was suspected that some of the basic postulates must be in error. For these general reasons the studies reported herein were begun. It was hoped thereby to obtain a picture of the mechanism by which CO_2, entering condensers with steam, becomes dissolved in the condensate.

GENERAL FINDINGS

The general findings are that theory (Henry's and Dalton's Laws) accurately defines the factors controlling the CO_2 content of condensates, and that the apparent anomalies between theory and field findings have resulted from an imperfect application of the laws. It is shown that the mechanics of condensation, as employed in common types of heating equipment, favor the production of condensates containing a high percentage of the CO_2 entrained with the steam; and that there is a progressive pick-up of CO_2 as the condensate moves down the return system.

THE CORROSIVITY OF CO_2-BEARING CONDENSATES

In an experimental set-up the potential corrosivity of CO_2-bearing condensates, at about 200 F, has been shown [1] to subscribe to the equation:

$$P = 0.46W^{0.53}$$

† Results of research studies in The Detroit Edison Co.
* Chemical engineer. Member of A.S.H.V.E.
[1] *Power Plant Engineering* (L. F. Collins), January, 1944.

Presented at the 51st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Boston, Mass., January, 1945.

where

P = corrosion rate as average penetration in inches per year × 1000. (Values of 20 or more are believed to indicate serious corrosion).

W = pounds of dissolved CO_2 contacting the metal surface in one hour × 100,000. (Equals CO_2 content of condensate in ppm × pounds of condensate flowing per hour × 0.1).

If this, or a comparable relationship, holds in actual practice, then it is clear that even low concentrations of dissolved CO_2 are capable of causing corro-

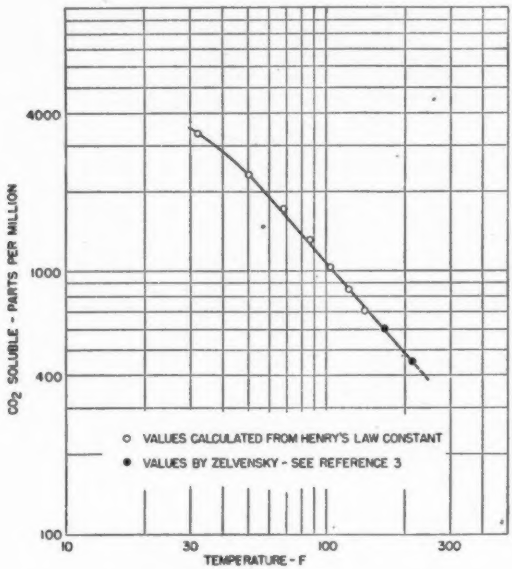

FIG. 1. SPECIFIC SOLUBILITY OF CARBON DIOXIDE IN OTHERWISE PURE WATER AT A PARTIAL PRESSURE OF 14.696 PSI ABSOLUTE

sion troubles. Thus, a complete knowledge of the mechanism which controls the solution of CO_2 by condensate should be helpful in designing corrosion control expedients.

FACTORS DETERMINING THE CO_2 CONTENT OF CONDENSATES AS PREDICTED BY THEORY

Theory holds that for any gas which does not react with water, the amount absorbed by a unit quantity of water, under conditions of equilibrium, decreases with increased temperature and is directly proportional to the pressure exerted by the gas upon the exposed surface of the solution.

A corollary states that when a mixture of gases is in contact with water, at a given temperature, there will be dissolved an amount of each gas corresponding to that part of the total gas pressure each exerts upon the water.

The rate of hydration of carbon dioxide to form carbonic acid has been measured, most recently, by Mills and Urey,[2] who also summarize the work of early investigators. These studies show that the rate of hydration is im-

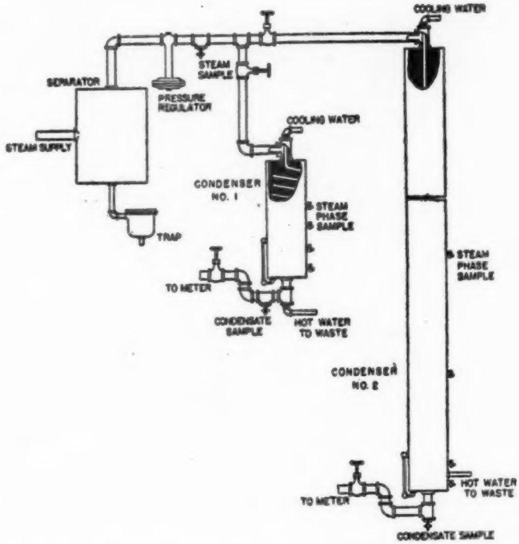

FIG. 2. ARRANGEMENT OF EXPERIMENTAL CONDENSERS

measurably fast, especially at the boiling point of water, but that less than one per cent of the total CO_2 present is combined with water as carbonic acid.

The specific solubility of CO_2 in water has been studied by a large number of investigators, most recently by Zelvensky,[3] who critically reviews most of the published data. He shows that, for low CO_2 concentrations, calculations based upon Henry's Law are valid. The curve in Fig. 1 shows the results of such calculation when the pressure of CO_2 in the vapor phase is one atmosphere. The values from 32 to 140 F are based upon a plot of Henry's Law constant[4] and those from 140 to 212 F upon the data of Zelvensky,[5] Bohr and Bock.[6] The extrapolation, shown by the broken line portion of the curve,

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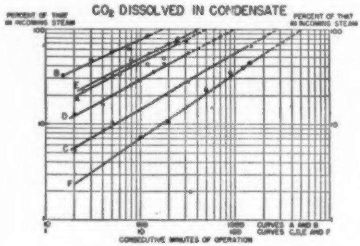
[2] *Journal American Chemical Society* (G. A. Mills and H. C. Urey), Vol. 62, 1940, p. 1019.
[3] *Journal Chemical Industry* (USSR—Y. D. Zelvensky), Vol. 1, 1937, p. 1250.
[4] International Critical Tables, Vol. 3, p. 260 (McGraw-Hill Book Co., Inc.).
[5] Loc. Cit. Note 3.
[6] Physico-Chemical Tables (Bohr & Bock), Vol. 2, p. 996, Lippincott, 1911.

has been made to cover the higher temperatures commonly found in steam heating equipment.

All of these data indicate that the amount of CO_2 in solution in the condensate flowing from any heating equipment will be governed principally by the CO_2 concentration of the steam in the vapor space of the equipment.

The CO_2 content of steams, used for heating purposes, usually is below 50 ppm, although occasionally values of the order of 100 ppm are encountered.



KEY

CURVE	A	B	C	D	E	F
CONDENSER	1	1	2	2	2	3
CONDENSING RATE lb per hr at 5 lb gage	3	7	100	200	200	55
CO_2 IN INCOMING STEAM parts per million by weight	18	18	38	23	40	40
TOTAL CO_2 TO SYSTEM lb per hr per cu in of steam space x 10⁶	18	43	75	90	157	24

FIG. 3. PROGRESSIVE INCREASE WITH TIME OF CO_2 CONTENT OF CONDENSATES

If the mechanics of condensation were such as to cause CO_2 values of this order (up to 100 ppm) to persist in the vapor space of steam condensing apparatus, calculations based upon Henry's Law show that immeasurably small amounts [7] of CO_2 will be dissolved in the condensate. Nevertheless, it has been shown [8, 9] that readily measurable amounts are often found. The first aim of these studies was to resolve these apparent contradictions.

TEST TECHNIQUE

Two water-cooled condensers, arranged as shown in Fig. 2, were used as experimental units. A *separator* was provided to prevent entrainment of boiler water salines, some of which interfere with the measurement of CO_2. The cooling water was made to run parallel with the flow of steam to prevent

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[7] Dissolved CO_2 concentrations of less than one part per million cannot be accurately determined with the methods commonly used.

[8] Some Fundamental Considerations of Corrosion in Steam and Condensate Lines, by R. E. Hall and A. R. Mumford. (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 121.)

[9] Corrosion in Steam Heating Systems, by L. F. Collins with E. L. Henderson. (*Heating, Piping and Air Conditioning*, September, 1939-May, 1940, inclusive.)

undercooling of the condensate. The rate of condensation was controlled by regulating the flow of cooling water. Steam pressure was automatically maintained at 5 psi gage by means of a reducing valve. Provisions were made for sampling the incoming steam, the vapor at various elevations in the steam space of the condenser, and of the condensate leaving the units. Escape of

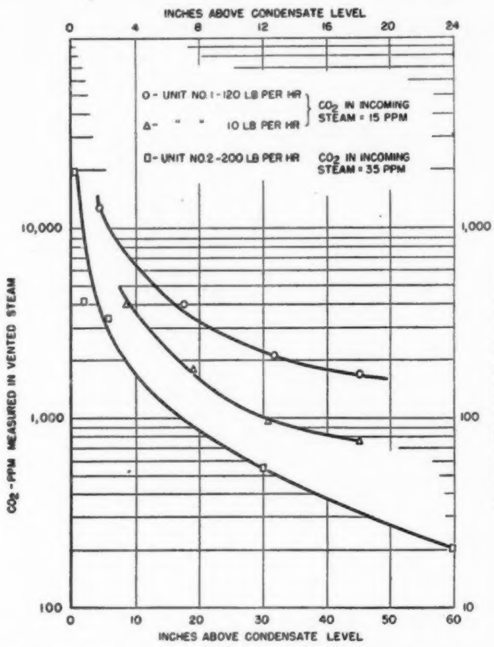

FIG. 4. CO_2 CONTENT OF VAPOR PHASE AT DIFFERENT HEIGHTS ABOVE CONDENSATE LEVEL

undissolved gas, along with the condensate, was prevented by throttling the discharge line so as to keep condensate always visible in the gage glass.

Time-Rate of Solution of CO_2

When these units were operated at different condensating rates, and with different amounts of CO_2 present in the incoming steam, it was found that, in a comparatively short period of time, the condensate contained an amount of CO_2 equal to that in the incoming steam. The progressive increase in the CO_2 content of the condensates with time is plotted in Fig. 3.[10] In view of

[10] Curve F was obtained from a very large cast-iron radiator used for other studies hereinafter reported.

the theory described previously, it was reasoned that for complete solution to occur the CO_2 content of the steam in the vapor space of the condensers must be much higher than in the incoming steam. This led to a study of the vapor phase at various levels.

CO_2 Content of the Vapor Phase at Different Levels

Repeated analyses of samples drawn from the vapor phase at different levels gave results typified by the curves in Fig. 4. These data are believed to pro-

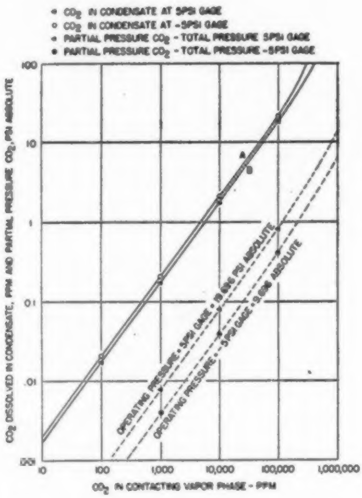

FIG. 5. RELATION OF CO_2 DISSOLVED IN CONDENSATE AND PARTIAL PRESSURE OF CO_2 IN CONTACTING VAPOR PHASE

vide a satisfactory explanation to the apparent contradiction between pure theory and actual operating conditions. They show clearly that in the vapor space steam and CO_2 do not form a homogeneous mixture of the same composition as in the incoming steam, as has been postulated,[11] but rather a gradient mixture several times richer in CO_2 than the incoming steam with the highest CO_2 concentration persisting at the *end of the steam path.*

In the test units, as in most designs of heating equipment, the end of the steam path was at the bottom. The curves in Fig. 4 indicate that at the condensate-gas interface the CO_2 concentration of the steam was several thousand parts per million. Curves A and B of Fig. 5, calculated upon the basis of Henry's Law, encompass the pressure range common to steam heating equip-

[11] Loc. Cit. Note 8.

ment. They show that CO_2 concentrations of the order indicated by Fig. 4 had to persist to bring about solution of CO_2 in amounts shown by the curves in Fig. 3. It should be noted, too, that lower CO_2 values in the steam phase are required when *undercooling* of the condensate occurs, as in certain types of water heaters.

Experimental difficulties precluded examination of the vapor layer closer to the condensate than the value shown in Fig. 4, *i.e.*, about 2 in. When attempts to do so were made, condensate was entrained with the vented steam.

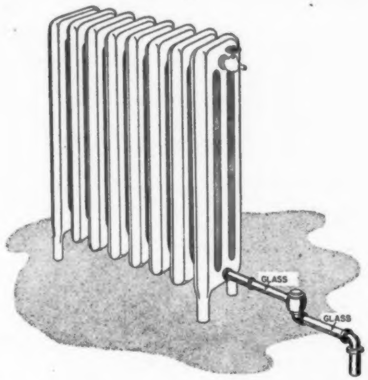

FIG. 6. ARRANGEMENT OF RADIATOR FOR THERMOSTATIC TRAP VENT STUDIES

In view of the fact that CO_2 concentrates at the end of the steam path, two questions present themselves:

1. Why not vent off the gas to the atmosphere, thus reducing proportionately the amount dissolved in the condensate?
2. Why is it that since conventional designs of equipment are seldom, if ever, provided with a venting device for removal of CO_2 and while measurable amounts of CO_2 are commonly found in solution in the condensate leaving such equipment very seldom, is it equal to that in the incoming steam?

Detailed studies covering the first question have been made and the results reported elsewhere.[12] Satisfactory answer to the second question, it is believed, has been obtained during the present studies.

ACCIDENTAL VENTING AND SOLUTION IN RETURN LINES

If a condensing unit, using CO_2-bearing steam, operates continuously, and the CO_2 dissolved in the outgoing condensate remains consistently less than in

[12] Preventing the Solution of CO_2 in Condensates by Venting of the Vapor Space of Steam Heating Equipment, by D. S. McKinney, J. J. McGovern, C. W. Young, and L. F. Collins (see p. 53 this volume).

the incoming steam, and if the general principles previously established herein are valid, then it follows that *accidental* venting must take place. If this occurs through the trap into the return line, two interesting questions are posed:

1. Do conventional types of thermostatic traps permit the escape of undissolved non-condensable gases?
2. What becomes of the undissolved CO_2 as it moves down the return line?

Venting Through Thermostatic Traps

To study the escape of undissolved gas through different makes of thermostatic traps, a cast-iron radiator with a rated capacity of slightly more than

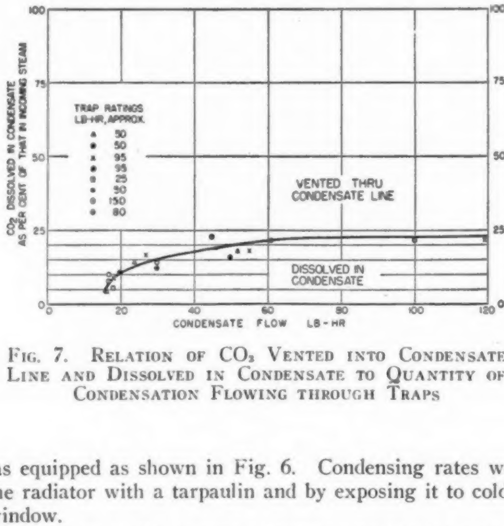

FIG. 7. RELATION OF CO_2 VENTED INTO CONDENSATE LINE AND DISSOLVED IN CONDENSATE TO QUANTITY OF CONDENSATION FLOWING THROUGH TRAPS

200 sq ft was equipped as shown in Fig. 6. Condensing rates were varied by blanketing the radiator with a tarpaulin and by exposing it to cold air admitted through a window.

It was found that when the amount of condensate flowing was not in excess of the rated capacity of the traps, the amount of gas vented into the return line and that dissolved in the condensate was as shown by the plot in Fig. 7.

The movement of non-condensable gas, occluded as bubbles, through the glass return line preceding the thermostatic trap was recorded, and a few of the pictures attesting this fact are shown in Fig. 8. That the bubbles were gas and not steam in attested by the facts (1) that their impingement upon the bellows caused no reaction, and (2) the condensate in the return line surged violently when steam bubbles were present.

Solution of CO_2 in Return Lines

For these studies, the specially designed return line, shown in Fig. 9, was used. The *loops* served as wells from which samples of condensate could be

withdrawn without entraining undissolved gas, and the upper half of the loops provided a path for movement of undissolved gases—and for steam when the set-up was operated as a one-pipe system.

One-Pipe System: Such systems usually operate within a pressure range of from slightly above atmospheric to 5 psi gage. The accumulations of CO_2 in the vapor space necessary to cause complete solution of CO_2, in amounts equal to that in the incoming steam (*i.e.*, up to 50 ppm), depress the tempera-

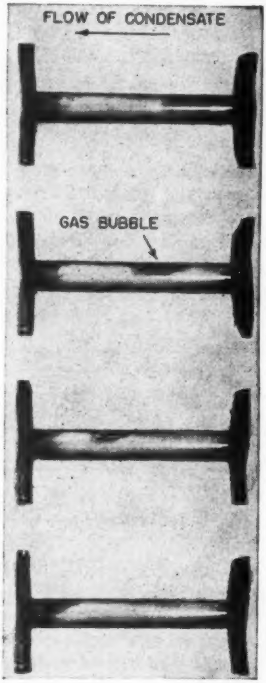

FIG. 8. PHOTOGRAPHIC EVIDENCE OF OCCLUSION OF UNDISSOLVED GAS IN CONDENSATE PASSING THROUGH THERMOSTATIC TRAP

ture of the resulting steam-CO_2 mixture a negligible amount as compared to pure saturated steam at the same total pressure. This is attested by the curves in Fig. 10. Thus, despite the fact that condensers on one-pipe systems are always provided with a thermostatically operated air vent valve, they are in reality unvented insofar as CO_2 is concerned.[13]

[13] Thermostatic air vent valves open at temperatures about 30 F below the temperature of saturated steam and close at temperatures about 20 F below that of saturated steam. (Private communication from a manufacturer.)

TABLE 1—CO_2 SOLUBILITY DATA—ONE-PIPE SYSTEM

Consecutive Hours of Operation	Condensing Rate Lb per Hr	Per Cent of Incoming CO_2[a] Dissolved in Condensate			
		Leaving Radiator	After Traversing Supply Line for		
			5 Ft	10 Ft	15 Ft
0.5	50	8	b	b	b
16	50	47	14	13	10
25	48	88	32	24	20
72	42	220	120	93	53

[a] CO_2 in Incoming Steam = 29 ppm.
[b] Unable to Obtain Satisfactory Sample.

Because of these facts, the radiator in these tests was not provided with a vent. The data in Table 1 show the gradual increase in dissolved CO_2 measured at key points over an operating period of 72 consecutive hours. The test was discontinued at the end of 72 hours because the radiator became partly *gas bound.* That the gas binding was due to air, not CO_2, is attested by an analysis of the accumulated non-condensable gas which gave the following results:

	Per Cent
Carbon dioxide	6.8
Oxygen	9.7
Hydrogen	4.3
Carbon monoxide	0.2
Nitrogen	79.0

The presence of oxygen and nitrogen is explained by the fact that the boiler feedwater, from which the steam was derived, was not completely deaerated.

Vacuum Return System: When operated as a vacuum return system—*i.e.*, with the radiator at a *positive* pressure and the return line under *vacuum*—the results typified by the data in Table 2 were obtained.

Each of the values shown in Table 2 is an average of conditions which persisted for periods of from five to ten consecutive days.

From these data it is clear that in vacuum return lines the condensate progressively dissolves CO_2 as it flows along, due apparently to decreasing condensate temperatures; but that the percentage increase is less as the vacuum on the lines reaches the higher values. It is equally clear that even a high vacuum does not guarantee a CO_2-free condensate where a CO_2-bearing steam is used.

A reason for the decrease in the CO_2 resulting from the passage of condensate through the vacuum pump and meter, where it comes into contact with the atmosphere, is postulated in the Appendix.

Vacuum System: When operating with both the radiator and return line at negative pressures, results typified by the data in Table 3 were obtained.

TABLE 2—CO_2 SOLUBILITY DATA—VACUUM RETURN SYSTEM

Condensing Rate Lb-Hr	CO_2 in Steam ppm	Per Cent of Incoming CO_2 Dissolved in Condensate: Leaving Radiator[a]	After Traversing Return Line for 5 Ft	10 Ft	15 Ft	Leaving Meter[b]	Vacuum on Return Line[c]
55	34	6	9	23	52	21	4 to 8
88	32	11	12	26	37	13	8 to 12
110	38	84	20	28	29	9	12 to 20

[a] Radiator Operating Pressure: 5 psi gage.
[b] Atmospheric Pressure.
[c] In. Hg.

The data in Table 3 are the average of conditions persisting for periods of from five to ten days.

A comparison of the data of Tables 2 and 3 show that with the experimental condensers operating under a slight vacuum more CO_2 is dissolved than under

TABLE 3—CO_2 SOLUBILITY DATA—VACUUM SYSTEM

Condensing Rate Lb-Hr	CO_2 in Steam ppm	Per Cent of Incoming CO_2 Dissolved in Condensate: Leaving Radiator	After Traversing Return Line for 5 Ft	10 Ft	15 Ft	Vacuum In. Hg: Leaving Meter[a]	On Radiator	Return Line
42	27	70	70	68	67	11	2 to 4	12 to 20
72	35	77	77	75	71	17	2 to 4	4 to 8

[a] Atmospheric Pressure.

a positive pressure, all other conditions being equal. This, it is believed, results from the lower condensate temperatures under vacuum conditions. It is also worth noting that, when large amounts of CO_2 are dissolved (note the condensation rate of 110 lb/hr—Table 2), flashing of the condensate into a

TABLE 4—CO_2 SOLUBILITY DATA—GRAVITY RETURN SYSTEM

Condensing Rate Lb-Hr	CO_2 in Steam ppm	Per Cent of Incoming CO_2 Dissolved in Condensate: Leaving Radiator	After Traversing Return Line for 5 Ft	10 Ft	15 Ft	Leaving Meter	Operating Pressure Psi-Gage: Radiator	Return Line
55	34	6	29	65	100	..	4	0
90	35	80	74	80	83	45	4	0

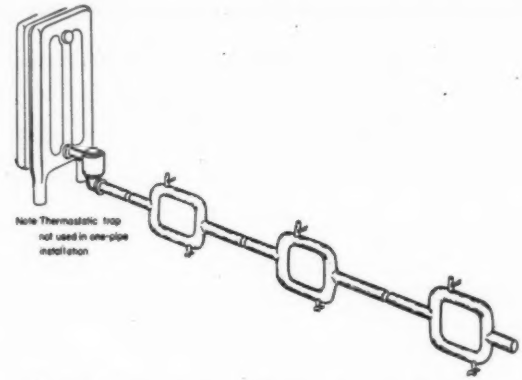

FIG. 9. ARRANGEMENT OF RETURN LINE IN RESOLUTION STUDIES

vacuum return causes a considerable decrease in the dissolved CO_2 content, whereas (Table 3) when little or no flashing occurs through the thermostatic trap, there is no release of CO_2.

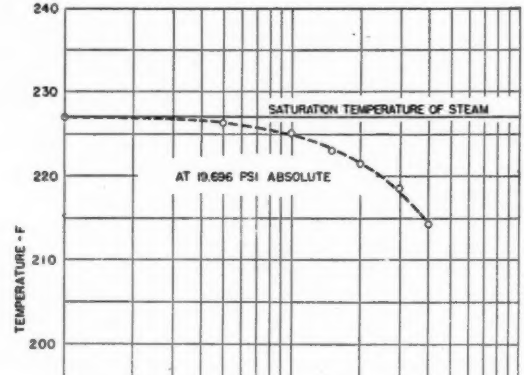

FIG. 10. TEMPERATURE PRESSURE RELATIONSHIP FOR CO_2-STEAM MIXTURE

Gravity Return System: When the radiator was operated at a positive pressure and the return line at atmospheric pressure, results typified by the data in Table 4 were obtained.

The data in Table 4 show that as the condensate moved down the return line there was a progressive pick-up of CO_2 until the condensate contains an amount equal to that in the incoming steam. Whether this resulted from decreasing condensate temperature or the accumulation of CO_2 in the vapor space of the return line (in the manner previously described for condensing

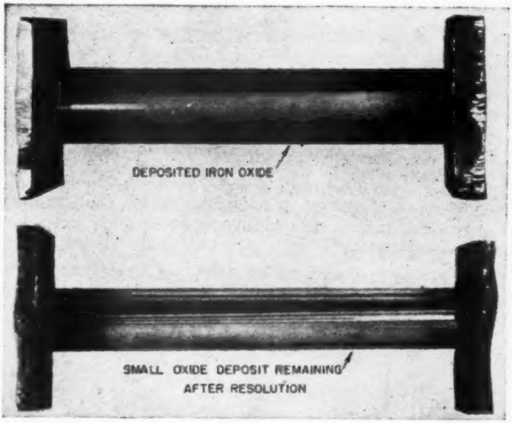

FIG. 11. PHOTOGRAPHIC EVIDENCE OF FORMATION AND RESOLUTION OF FERRIC IRON IN A RETURN LINE

apparatus) or for some other reason is not clear at this time. An interesting observation is the decrease in CO_2, at the higher condensing rate, produced by the condensate flashing as it passed the thermostatic trap. The same thing is shown, for the highest condensation rate, in Table 2.

APPENDIX

CONCERNING INFLUENCE OF IRON REACTION PRODUCTS

Plugging of the return line immediately following thermostatic traps frequently is one of the chief causes of corrosion troubles. Analyses of a large number of such deposits have shown them to consist of iron reaction products. Invariably, the outer crust is of ferric iron, the layer adjacent to the metal is of ferrous iron, and the intermediate strata is of magnetic oxide. The author has postulated, on occasions, that these deposits result from (1) the solution of iron in the condensate in the condenser which (2) quickly oxidize after entering the return line where, invariably, air is present. In the present studies an opportunity was afforded to obtain proof of these reactions.

Precipitation and Resolution of Iron Reaction Products

In the thermostatic trap vent studies, wherein the glass return lines were available, no rust deposits were ever found preceding the trap. However, when the system was operated so that the line following the trap was about half full of condensate, and air could, therefore, seep back to the trap, a generous layer of red oxide would be formed in two or three days. If, subsequently, the condensation rate was raised so as to cause the return line to stay full of condensate, no deposit formed and that already deposited would redissolve. The pictures in Fig. 11 show the return line after the deposit had formed and the same line after resolution of most of the deposit.

Indices of the Influence of Ferrous Iron on the Solubility of CO_2

In Tables 2, 3, and 4 of the main text it is seen that in every instance there was a decided decrease in the CO_2 content of the condensate after it had passed through the condensation meter. Upon passing the latter it came into contact with the atmosphere. Samples drawn preceding the meter were always water-white, while those after the meter always contained precipitated ferric iron. It is felt that the presence of ferrous iron materially influences the solubility of CO_2 in such solutions. This would explain the sudden loss of CO_2 by the condensate upon passing through the condensate meter, where the ferrous iron was quickly oxidized to the ferric state.

Check analysis of iron-laden condensate samples by the evolution method as against direct titration with standard sodium bicarbonate, using phenolphthalein, failed to show that the iron reaction products fouled the direct titration method.

A few analyses of iron-laden condensate gave the results shown below:

pH VALUE[a]	CO_2 PPM[b]	TOTAL IRON AS PPM Fe_2O_3[c]
5.5	2	0.8
5.5	8	1.3
5.7	2	0.8
5.7	27	4.6
5.9	11	5.5
5.9	36	24.5
6.2	27	17.0
6.2	16	36.8
4.8	135	36.1

[a] Determined electrometrically using glass electrode.
[b] By direct titration with Na_2CO_3 using phenolphthalein.
[c] Determined gravimetrically.

Whether these data have significance is not known. It is interesting to speculate, nevertheless, on the possibility of reducing CO_2 attack by oxygenation of the condensate.

(See p. 68 for Discussion)

No. 1266

PREVENTING THE SOLUTION OF CO_2 IN CONDENSATES BY VENTING OF THE VAPOR SPACE OF STEAM HEATING EQUIPMENT

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This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with Carnegie Institute of Technology.

THIS PAPER presents the results of cooperative research studies [1] aimed at revealing, within practical limits, the possibilities of preventing the solution of CO_2 in condensates by venting of the vapor space of steam heating equipment. This work was undertaken in an attempt to discover another means of minimizing corrosion of condensate return lines when, perforce, CO_2-bearing steams must be used. It was predicated upon the previously published findings [2] of one of the authors which show that, other conditions being fixed, corrosion of condensate lines decreases as the dissolved CO_2 content of the condensate decreases.

General Findings

The general findings are that venting of the vapor space of steam condensers does not provide a means of producing CO_2-free condensates even when steams containing small concentrations of the gas are used. Notwithstanding, it is possible to produce thereby, from steams *rich* in CO_2, condensates containing gas concentrations of the same order of magnitude (2 to 4 ppm) as ordinarily are formed in equipment using steams of the *lowest* CO_2 content (*i.e.*, about 5 ppm) that are now generated from carbonate-bearing feedwater upon a commercial scale. Theoretically, therefore, venting provides a means of minimizing but not entirely preventing corrosion. Its general use, however, must await the commercial availability of suitable venting devices.

General Considerations

In 1939, Collins [2] published the curves reproduced in Fig. 1. These show that, other conditions being fixed, corrosion is proportional to the CO_2 concentration of condensates. The work of Mills and Urey [3] and their predecessors show that, at heater temperatures, equilibrium of CO_2 between steam and con-

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[1] Cooperative Research at Carnegie Institute of Technology, September, 1942, to December, 1943, sponsored by the A.S.H.V.E. Technical Advisory Committee on Corrosion.
[2] Corrosion in Steam Heating Systems, by L. F. Collins and E. L. Henderson. (*Heating, Piping & Air Conditioning*, Sept. 1939 to May, 1940, incl.)
[3] *Journal American Chemical Society*, Vol. 62, p. 1019, 1940.

Presented at the 51st Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, Mass., January, 1945.

densate can be attained in an extremely short period of time. At equilibrium, the CO_2 content of any condensate, free of alkaline entrainment, will be directly proportional to the partial pressure of CO_2 in the vapor phase contacting the condensate. In his latest publication, Collins[4] shows that in conventional designs of steam heating equipment, using CO_2-bearing steam, stratification of CO_2 and steam occurs and that the concentrations of CO_2 which

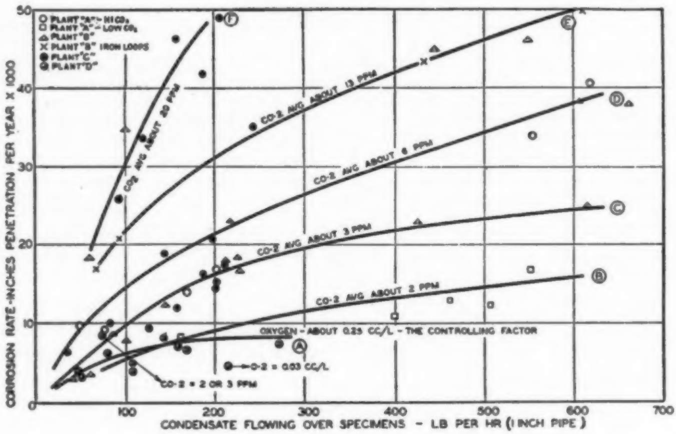

FIG. 1. CURVES SHOWING THE RELATION BETWEEN CORROSION AND GAS CONTENT OF CONDENSATE WHEN VARYING QUANTITIES OF CONDENSATE ARE FLOWING

eventuate may cause the hot condensate to dissolve an amount of CO_2 equal to that in the incoming steam.

From these observations it can be reasoned that, if CO_2 can be prevented from accumulating in the vapor space, lower partial pressures of CO_2 will result, less will dissolve in the condensate and, therefore, the latter will have a proportionately lower potential corrosivity. It was the aim of the present studies to determine the practical possibilities of venting gaseous CO_2 from the vapor space of heating equipment operating at high condensing rates, such as a water heater.

EXPERIMENTAL APPROACH

In the design of all heating equipment, the principal objective sought is to achieve the ultimate in thermal economy. Normally, steam and condensate are caused to flow in the same direction. In addition, in some units, wherein water

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[4] Studies of the Mechanism of Solution of CO_2 in Condensates Formed in the Steam Heating Systems of Buildings, by Leo F. Collins. (See p. 39, this volume.)

is the cooling medium, steam and condensate are caused to flow counter current to the cooling water as a result of which *undercooling* of the condensate occurs.

In view of the principles set forth in the preceding paragraph, it will be recognized that the design features cited favor solution of CO_2 in the condensate. If this is true, then it should follow that, if steam and condensate are made to flow in opposite directions and if the CO_2, which accumulates at the end of the steam path, is vented to the atmosphere, conditions favoring solution of a minimum of CO_2 in the condensate should be produced. The present studies were conducted in such a manner as to test these postulates.

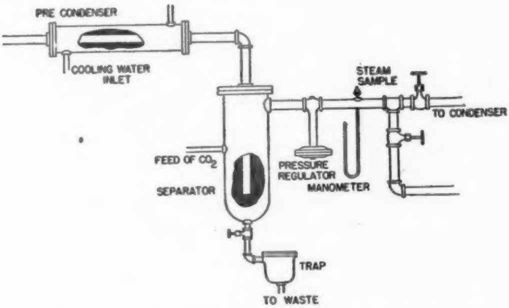

FIG. 2. ARRANGEMENT OF EQUIPMENT PRECEDING EXPERIMENTAL CONDENSERS

TEST TECHNIQUE

Steam from a main supplying the laboratory building was used throughout the tests. To minimize the interference of entrained boiler water salines, and to facilitate enrichment of the steam with CO_2 when desired, the equipment preceding the condensers was arranged as shown in Fig. 2.

In all tests, to prohibit the escape of undissolved gas along with the condensate, the condensate line was kept flooded at a predetermined level. This was accomplished automatically by means of a *home-made* constant-level float valve. Essentially, this device consisted of a glass float linked to a carefully machined brass cylinder. As the latter raised or lowered, it throttled the exit port in the condensate line.

Condenser No. 1

This unit was a water heater of the helical coil-type, made entirely of copper alloy. The exterior surface of the coil and the interior surface of the jacket were heavily tinned to eliminate the possible effects of corrosion products. It had a rated capacity of 180 gal per hour when utilizing steam at atmospheric

pressure, with an inlet water temperature of 40 F and an outlet water temperature of 140 F, *i.e.*, it contained about 3.5 sq ft of heating surface.

The first series of tests were made with steam and condensate flowing in the same direction and counter-current to the cooling water, *i.e.*, conventional operation. A steam pressure of 3 psi gage was maintained, and the cooling water was regulated so as to cause condensation of about 225 lb of steam per hour.

The vent and subsequent venting apparatus were arranged as shown in Fig. 3. With these arrangements, data were collected while venting steam at

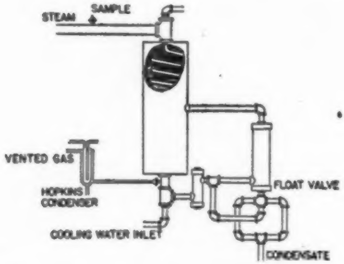

FIG. 3. ARRANGEMENT OF EXPERIMENTAL CONDENSER No. 1

different rates up to 1.81 per cent of that entering the heater, with the incoming steam containing about 4 ppm of CO_2 and also after it had been enriched to contain about 40 ppm of the gas.

The data collected are summarized in Table 1. The values shown are an average of 4 readings taken at consecutive 15-min intervals after the unit had operated sufficiently long, at a given venting rate, to reach a steady state.

TABLE 1—AVERAGE RESULTS OBTAINED WITH HEATER No. 1

Condensing Rate Lb per Hr	CO_2 ppm in			Venting Rate[a]	Per Cent of Incoming CO_2 Measured in	
	Incoming Steam	Vent Steam	Condensate		Condensate	Vented Steam
238	3.6	134	1.7	1.73	42.6	57.4
222	3.0	163	1.4	1.25	45.7	54.3
241	3.8	252	2.3	0.90	49.0	51.0
226	3.3	407	1.8	0.43	47.2	52.9
232	39	1886	6.2	1.81	15.7	84.3
232	44	3500	8.2	0.98	19.3	80.7
225	37	3300	6.1	0.94	16.6	83.4
238	33	5290	8.0	0.52	23.9	76.1
236	44	6940	11.1	0.43	27.1	72.9

[a] As per cent of incoming steam.

When an attempt was made to operate this unit with counter-flow of steam and condensate, it was found impossible to condense as much steam as in the first series of tests. Because of this experimental difficulty it was decided to cease work with this unit and to begin the studies on Condenser No. 2.

Condenser No. 2

This unit was a water heater of the hair pin tube type. The tubes were of copper alloy, and the shell was of steel. Prior to starting the test, all the inside

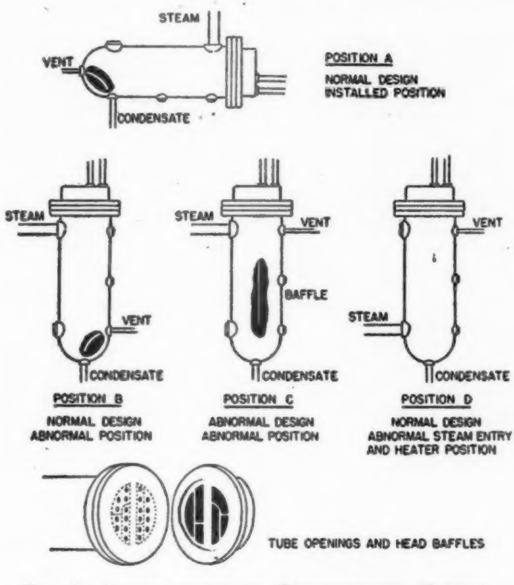

FIG. 4. ARRANGEMENTS OF EXPERIMENTAL CONDENSER No. 2

surfaces of the shell and the outside surfaces of the tubes were tin plated. This was done to avoid interference of corrosion products, which conceivably may affect the solubility of CO_2 in condensates so contaminated. It has been shown [5] that tin is practically immune to carbonic acid corrosion. This heater had a rated capacity of 400 gal per hour when utilizing steam at atmospheric pressure, with an inlet and outlet water temperature of 40 and 180 F respectively, *i.e.*, it contained 9 sq ft of heating surface.

In all the tests involving Unit No. 2, a vent-condenser [6] was used which differed in one respect from the design used with Unit No. 1. Why this

[5] More Information on Corrosion in Steam Heating Systems, by L. F. Collins. (Proceedings 4th American Water Conference *Engineers' Society of Western Pennsylvania*, November, 1943.)

[6] In all tests, a modified Hopkins Condenser was used as a vent condenser. This is a small glass condenser common to chemical laboratories.

change was made is described later in the paragraph entitled *Criteria for the Design of Venting Devices.*

With the heater positioned and/or arranged in the four different ways, depicted by the sketches in Fig. 4, tests were made at condensation rates corresponding to 300 and 600 lb of steam, approximately, per hour at 3 psi gage. For each condensation rate, tests were made at CO_2 levels in the incoming steam of 30 and 60 ppm respectively. For each CO_2 level, tests were run at rates of venting corresponding to 0.25, 0.50, and 1.0 per cent of the weight of steam entering the heater in unit time.

In every test, the following general procedure was used:

The heater was put into operation and adjustments were made to give the desired condensation rate. The CO_2 feed to the incoming steam was set at a predetermined value, as was also the venting rate. When the system had reached a steady state (usually in about one-half hour) as shown by preliminary checks, readings were taken. The data hereinafter presented represent the average of the values obtained from three or more consecutive readings made at about 20-min intervals. Thereafter, the venting rate was changed to a lower predetermined value and the same procedure repeated.

Normal Design and Installed Position—(A): With the unit installed in a horizontal position (arrangement A, Fig. 4), *which is its normal position in actual practice,* it seems permissible to assume that, as the condensate from the upper elevation gravitates downward, it is likely to contact cooler tubes. This, coupled with the fact that at the lowest level the gas phase richest in CO_2 persists, should make for maximum solution of the gas in the condensate. The heater in question was provided with a vent, as shown by arrangement A of Fig. 4 and tested under these conditions. The average results obtained are given in Table 2.

Normal Design—Abnormal Position—(B): In this series the heater's original design was not modified, but the unit was positioned so the tubes were vertical, *i.e.,* arrangement B of Fig. 4. With such an arrangement, it seems improbable that undercooling of the condensate can take place. Under these conditions, the average results obtained are those given in Table 3.

TABLE 2—AVERAGE RESULTS—HEATER NO. 2—NORMAL DESIGN AND INSTALLED POSITION

Condensing Rate Lb per Hr	CO_2 ppm in			Venting Rate[a]	Per Cent of Incoming CO_2 Measured in	
	Incoming Steam	Vent Steam	Condensate		Condensate	Vented Steam
630	61	4035	13	1.08	21.4	76.9
630	56	7550	17	0.50	30.4	69.0
606	55	11158	23	0.27	42.0	56.0
614	34	2108	11	1.11	32.4	69.1
614	33	3357	14	0.54	43.0	56.7
612	33	5894	15	0.25	45.4	54.6
366	59	4728	15	0.95	25.4	74.9
366	58	12148	17	0.35	28.6	72.3
366	58	18420	17	0.24	29.6	70.8
370	29	2470	11	0.75	37.6	62.1
366	31	4605	10	0.45	31.0	67.8
366	29	9142	9	0.22	31.0	66.9

[a] As per cent of steam entering unit.

TABLE 3—AVERAGE RESULTS—UNIT NO. 2—NORMAL DESIGN—ABNORMAL POSITION

Condensing Rate Lb per Hr	CO_2 ppm in			Venting Rate[a]	Per Cent of Incoming CO_2 Measured in	
	Incoming Steam	Vent Steam	Condensate		Condensate	Vented Steam
575	56	3090	7	1.00	13.1	87.6
552	56	7324	9	0.70	15.8	84.5
552	60	10416	15	0.43	25.0	75.6
552	54	16078	22	0.17	40.6	55.9
578	30	2548	5	1.04	16.0	84.4
576	32	4664	9	0.61	24.2	75.5
576	39	7008	10	0.38	25.6	73.4
576	32	10748	13	0.19	41.2	59.0
315	55	4621	7	1.11	12.4	86.1
321	56	10757	12	0.42	20.6	80.3
315	54	13685	17	0.27	31.8	68.3
330	33	3130	7	0.93	20.9	79.9
330	31	4826	11	0.46	34.5	65.6
330	32	7485	14	0.24	43.6	56.6

[a] As per cent of steam entering unit.

Abnormal Design—Abnormal Position—(C): For this series, the heater's design was modified to the extent that a baffle was installed to direct the path of the steam, as shown in arrangement C of Fig. 4, and the vent was relocated, as is also shown in the sketch. By this arrangement, undercooling of the condensate was prevented, and the CO_2 which normally accumulates at the bottom was forced to the top of the unit. Under these conditions, the average results obtained are those given in Table 4.

TABLE 4—AVERAGE RESULTS—UNIT NO. 2—ABNORMAL DESIGN—ABNORMAL POSITION

Condensing Rate Lb per Hr	CO_2 ppm in			Venting Rate[a]	Per Cent of Incoming CO_2 Measured in	
	Incoming Steam	Vent Steam	Condensate		Condensate	Vented Steam
550	59	6150	2	1.15	3.5	97.1
555	59	10630	3	0.55	5.1	95.0
560	57	21172	4	0.29	6.9	93.0
606	36	2900	4	1.07	11.1	89.0
579	35	5995	5	0.54	14.2	86.0
598	35	11546	6	0.26	17.2	82.4
300	60	5741	4	1.05	6.7	93.1
300	59	11233	5	0.53	8.5	92.7
300	60	21710	6	0.24	10.0	89.3
316	30	2778	4	1.00	13.3	87.1
310	30	5422	4	0.51	13.3	87.2
310	31	9648	6	0.25	19.3	81.0

[a] As per cent of steam entering unit.

Normal Design—Abnormal Steam Entry and Heater Position—(D): In this series the steam was caused to enter the unit close to the normal condensate level, and the vent was located at the top of the unit, as shown in arrangement D of Fig. 4. This, in effect, was the same as the arrangement in the tests of series C but accomplished without installing the baffle. With this arrangement, the average results obtained are those given in Table 5.

VENTING EFFICIENCIES

A comparative study of the data from the five series of tests indicates clearly that control of the CO_2 content of condensates, formed from and in contact with CO_2-bearing steams, by venting of the vapor space is limited and conditioned by:

1. The amount of steam vented in unit time.
2. The CO_2 content of the incoming steam.
3. The proper location of the bleed point.
4. Design characteristics of the unit in question.

Optimum Percentage of Venting

There are well informed engineers who contend that the venting of steam heating equipment, as a means of thwarting corrosion, is predestined to be impractical. Their capital argument is that the problem of disposing of the vented steam will be as costly and involved as the replacement of those parts which normally fail due to corrosion.

It will be shown later that, conceivably, there are rather simple ways for disposing of small amounts of vented steam. It will also be shown that, when intelligently employed, venting of small amounts of steam effectively minimizes the CO_2 content of condensates. Accordingly, it is believed that the

TABLE 5—AVERAGE RESULTS—UNIT NO. 2—ABNORMAL STEAM ENTRY AND HEATER POSITION

Condensing Rate Lb per Hr	CO_2 PPM in			Venting Rate[a]	Per Cent of Incoming CO_2 Measured in	
	Incoming Steam	Vent Steam	Condensate		Condensate	Vented Steam
624	58	4863	4	1.01	6.9	92.5
614	57	9193	4	0.55	7.0	93.1
608	55	17328	6	0.27	10.9	89.1
612	31	2517	3	1.05	9.7	90.2
614	30	4700	3	0.53	10.0	89.8
612	27	8923	4	0.26	14.8	85.4
310	60	6136	3	1.02	5.0	95.3
310	61	11778	4	0.51	6.6	94.0
310	60	21788	5	0.27	8.3	92.0
328	28	2714	2	0.96	7.1	92.4
324	28	5227	3	0.48	10.7	89.3
324	30	10789	6	0.25	20.0	79.6

[a] As per cent of steam entering unit.

present studies outline a method for control of carbonic acid corrosion that heretofore has not been available for use by steam utilization engineers.

Prior to the inauguration of these studies it was decided, quite arbitrarily, that for venting to be practical, not more than about two per cent of the steam entering a given condenser could be so dissipated. Thus it is that all tests were made at venting rates below two per cent.

In Figs. 5, 6 and 7, inclusive, are plotted the amounts of CO_2 found in the condensates (as per cent of that in the incoming steam) against venting rates,

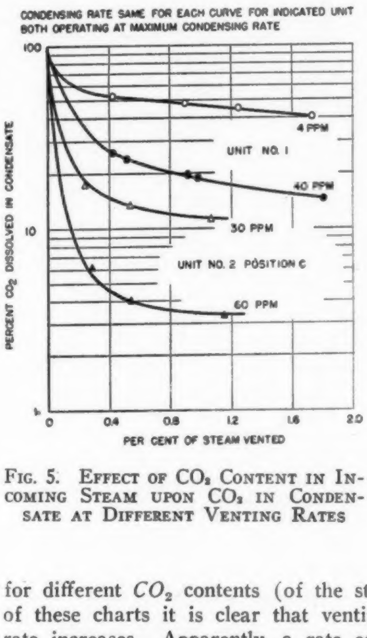

Fig. 5. Effect of CO_2 Content in Incoming Steam upon CO_2 in Condensate at Different Venting Rates

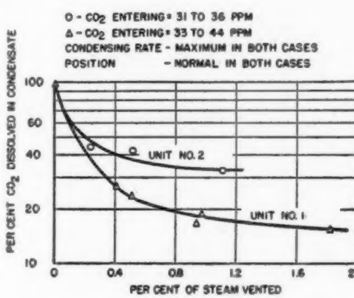

Fig. 6. Comparison of CO_2 Content in Condensate for Units 1 and 2 Operated at Maximum Condensation Rate with Steam of Comparable CO_2 Content

for different CO_2 contents (of the steam) and heater arrangements. In all of these charts it is clear that venting efficiency[7] decreases as the venting rate increases. Apparently, a rate equal to about one-half per cent of the steam entering a given unit is the most efficient, providing venting is done from the optimum bleed point.

In Figs. 5, 6 and 7, plotting the value of 100 per cent solution of the gas at zero venting rate is not based upon an academic postulate. Tests of both heaters showed that this occurs. When unit No. 1 had reached a steady state, the vent was suddenly closed. Under these conditions, complete solution of the gas was brought about in about 30 min. When unit No. 2 (arranged as shown by A of Fig. 4) had reached a steady state, at a venting rate of 1.35 per cent, closing of the vent brought about complete solution of the gas in about 20 min at a condensing rate of 600 lb of steam per hour.

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[7] Since the per cent of CO_2 removed is not a straight line function of the per cent of steam vented, doubling the venting rate does not halve the per cent CO_2 remaining in the condensate.

Effect of CO_2 Concentrations in Incoming Steam

Venting efficiency decreases as the CO_2 content of the incoming steam decreases. For CO_2 content of steam below about 4 ppm, venting is apparently of little practical value. The latter observation is attested by the curves in Fig. 5. It should be noted that the data for unit No. 2, plotted therein, which pertain to series C, are practically identical with the results obtained with series D.

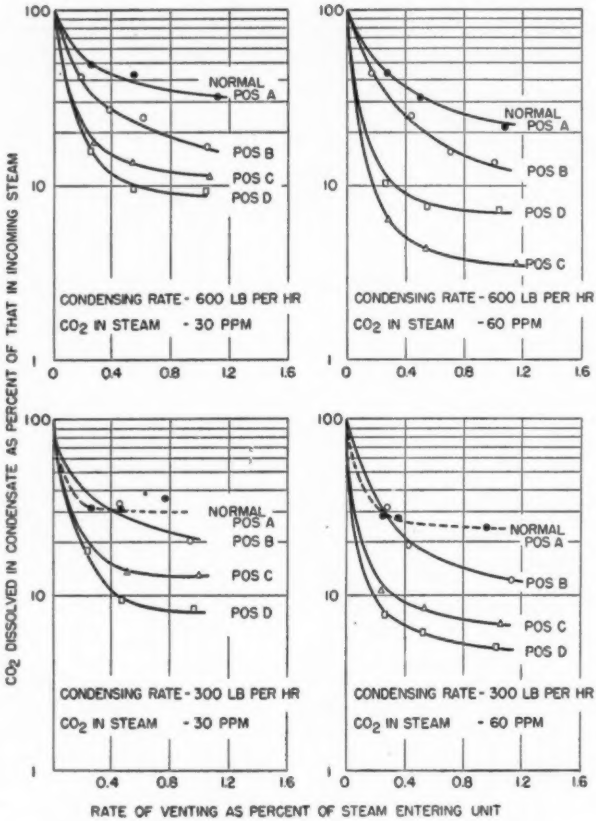

FIG. 7. EFFECT OF FOUR DIFFERENT ARRANGEMENTS ON UNIT No. 2 WHEN OPERATED AT TWO CONDENSATION RATES AND TWO CO_2 LEVELS

Optimum Bleed Point

Obviously, the optimum bleed point is at that location where the highest CO_2 concentration exists. While ordinarily this will be close to the condensate level, *i.e.*, in the bottom of the unit, it may be elsewhere, depending upon the direction and path of steam flow. The point of highest CO_2 concentration is always at the end of the steam path. It was for these reasons that in the tests of series A and B venting was from the bottom, whereas in series C and D it was from the top of the experimental unit.

Initially, there is a temptation to postulate that CO_2 accumulates at the bottom of conventional types of steam-condensing apparatus because of the differences in density between steam and CO_2. That such accumulations are due to steam flow rather than to differences in gas densities is attested by the findings of series A and B as contrasted with those of series C and D. The data in Tables 4 and 5 show higher CO_2 values for the vented steam than in Tables 2 and 3, all other conditions being equal. It is inconceivable that this could have occurred if gas densities were the controlling factor.

With the unit arranged as shown by B of Fig. 4, which is essentially an orthodox arrangement, the vent was suddenly closed so that the gas could accumulate. When equilibrium became established, *i.e.*, the amount of CO_2 dissolved in the outgoing condensate equalled that entering with the steam, sufficient heating surface had been blanketed with gas to reduce the steam condensing rate slightly over one per cent. With the arrangement shown by D of Fig. 4, which is an unorthodox arrangement, when equilibrium had become established, the steam condensing capacity had been decreased about 30 per cent. Certainly, therefore, an arrangement such as is shown in D is definitely impractical unless a suitable venting apparatus is provided.

Effects of Design Characteristics

The curves in Fig. 6 contrast the CO_2 dissolved in the condensate for the two units studied, when both were operated in an orthodox manner at maximum condensation rates and with steams of comparable CO_2 contents. The better performance of unit No. 1, it will be shown presently, resulted from the fact that undercooling of the condensate did not take place as is the case for the values relating to unit No. 2.

The curves in Fig. 7 show the average results obtained with unit No. 2 when it was arranged in the four different ways, depicted by the sketches in Fig. 4 but operated at two different condensation rates and two CO_2 levels for the incoming steam.

These charts demonstrate clearly (1) the effects of undercooling of the condensate, and (2) the value of utilizing steam scavenging.

In a unit of this design, arranged in its normal position, the condensate formed at the upper elevations is likely to contact progressively cooler tubes. This favors *undercooling* of the condensate and should become more pronounced as the rate of condensation increases. It has been shown, too, that with such an arrangement, the vapor in contact with the undercooled condensate is richest in CO_2; that both of these factors make for maximum solution of CO_2 under otherwise identical conditions. That this occurred is clearly shown by the data plotted in Fig. 7. That undercooling takes place more at high condensation rates is similarly shown.

The effects of undercooling can be demonstrated, too, in another manner. By comparing arrangements A and B it can be visualized that in arrangement B the condensate drained quickly from the vertical tubes, while in arrangement A it flowed from one tube to another. Thus it should follow that other conditions being equal, a given CO_2 content in the gas phase should bring about less solution of CO_2 in the condensate with arrangement B than with arrangement A. The curves in Fig. 8 show the relationship found to exist. It is interesting to note that the values for unit No. 1 subscribe to the lower curve. Apparently in this unit undercooling did not take place.

In comparing the results obtained by the two methods of steam scavenging it is seen that, except in one series, bringing the *purest* steam into contact

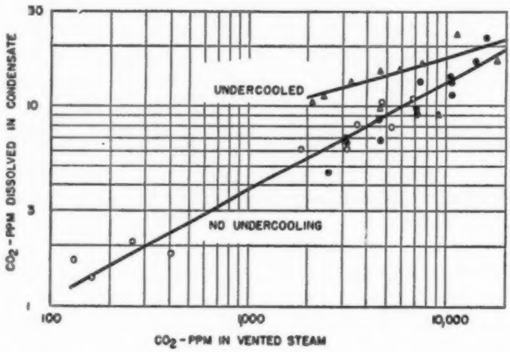

FIG. 8. EFFECT OF UNDERCOOLING UPON CO_2 DISSOLVED IN CONDENSATE

with the outgoing condensate is the most effective. The exception, however, focuses attention upon an important point, namely, the necessity for causing the steam to follow a predetermined path which encompasses the whole area of the condensate. Because of the high condensation rate used in the tests of series D and the high CO_2 in the incoming steam it is believed a *secondary pocket* of CO_2 accumulated, thus curtailing the effectiveness of the scavenging steam. This same phenomenon has been experienced in early design of deaerating types of industrial boiler feedwater heaters.

CRITERIA FOR THE DESIGN OF VENTING DEVICES

Practically, venting is impossible with equipment operating at subatmospheric pressures. On the other hand, a system in which the condensing equipment operates at positive pressures and the return lines under negative pressures would appear to yield most readily to the employment of venting.

In building steam heating equipment it rarely happens that a single piece of equipment is found which condenses more than about 1000 lb of steam per hour. If venting of about one-half per cent of the incoming steam represents

the optimum venting rate, then disposal of not more than five pounds of condensate per hour comprises the mechanics of the problem of venting.

To be practical, the venting device should separate the noncondensable gases from the condensate formed in the venting device and purge them to the atmosphere; returning the condensate, containing a negligible amount of dissolved gas, to the main return line.

Throughout the early experiments with unit No. 1, the modified Hopkins condenser was constructed as shown by arrangement A of Fig. 9. With this design the vented condensate immediately contacted the *cold finger* and tended to *hold* CO_2. For this reason the condenser was further modified as shown in arrangement B of Fig. 9. In this design the condensate in the venting

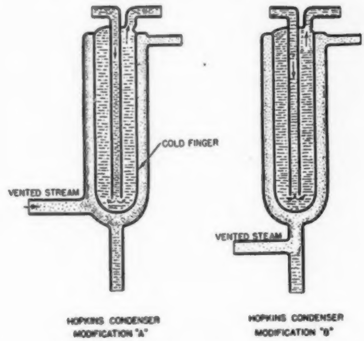

FIG. 9. CONSTRUCTION OF MODIFIED HOPKINS CONDENSER USED IN EARLY EXPERIMENTS WITH UNIT NO. 1

apparatus is continuously *boiled* by the incoming steam. The gas content of the condensate leaving both types of apparatus, under otherwise identical conditions, is typified by the data in Table 6. Clearly, the modified design B was a distinct improvement.

It is believed that condensate of the quality typified by the data for design B may safely be returned to the main return line. While its CO_2 content will be comparatively high, the amount of condensate produced is so small that the

TABLE 6—DATA COMPARING THE PERFORMANCE OF TWO MODIFIED HOPKINS CONDENSERS

STEAM CONDENSED LB PER HR	MODIFICATION A CO_2 PPM IN STEAM TO CONDENSER	CONDENSATE LEAVING CONDENSER	STEAM CONDENSED LB PER HR	MODIFICATION B CO_2 PPM IN STEAM TO CONDENSER	CONDENSATE LEAVING CONDENSER
4.2	1886	12	..	...	...
2.8	3500	35	3.1	2714	28
1.2	5290	375	1.6	5227	100
1.0	6940	967	0.8	10789	156

increase in CO_2 concentration of the aggregate in the main return line will not be of practical importance.

In the design of a venting device which utilizes water as the cooling medium, provision should be made to guarantee no undercooling of the condensate. A device made of finned-type tubing, thus using air as the cooling medium, would appear preferable for most services. In Fig. 10 a possible design is suggested.

(1) Steam inlet
(2) Steam outlet orifices
(3) Tubing ⅛ in. diameter
(4) Condensate outlet-orificed-to main return or waste
(5) Threaded connection
(6) Splash plate
(7) Fluted tubing
(8) Sludge pocket
(9) Shield
(10) Outlet for gases

Materials of Construction: 18–8 stainless steel, aluminum, or tin plated copper alloys. (Same material should be used throughout.)

Note: Unit must be mounted in vertical position. Not usable on condensers operated at negative pressures.

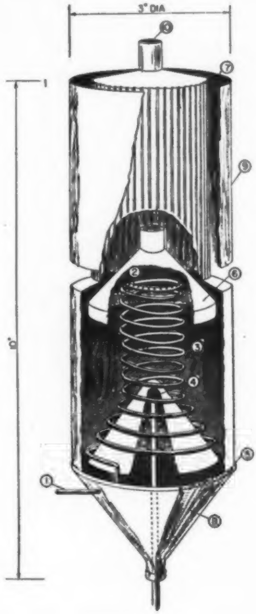

FIG. 10. ONE POSSIBLE DESIGN OF AN AIR-COOLED VENT CONDENSER

Insofar as the authors can determine few, if any, of the so-called air vent valves operated thermostatically, appear suitable for venting CO_2. The apparent reason is that the accumulation of CO_2 necessary to induce solution of important quantities of CO_2 does not depress the temperature of the steam-gas mixture sufficiently to cause the valves to open.

SUMMARY

It has been shown that by intelligent venting of the vapor space of steam condensing equipment, steams rich in CO_2 can be caused to produce condensates containing small amounts of CO_2, i.e., of the order of 2 to 4 ppm; and that venting is of little practical value when the incoming steam contains less than

about 5 ppm of CO_2. It has been shown elsewhere[8] that where boilers must use important quantities of carbonate-bearing feedwater, the inadequacy of water processing methods now in use preclude the commercial production of steam containing less than such quantities of CO_2; and that CO_2 concentrations in the condensate of the order of 2 to 4 ppm are capable of causing corrosion trouble if other contributing factors are optimum.

In view of these facts, it is clear that venting provides a means of mitigating but not entirely preventing corrosion troubles due to carbonic acid. As such, it is an expedient which lies definitely within the province of the steam utilization engineer and its effectiveness is comparable to the water processing methods available to engineers in charge of steam production.

Managements should appreciate that, with the means now available, the efforts of both the steam utilization engineer and the steam production engineer are limited to minimizing rather than to preventing corrosion when perforce the steam generator uses important quantities of carbonate-bearing feedwaters.

Apparently the ultimate solution of the CO_2 corrosion problem must await the development of suitable chemicals which can be added to the steam to neutralize carbonic acid, or the development of methods for processing carbonate-bearing feedwaters so as to completely eliminate the entrainment of CO_2 with the steam. Studies dealing with neutralizing chemicals, especially those of an organic nature, are in progress elsewhere.[9] Studies dealing with the development of improved water processing methods are under consideration by the A.S.H.V.E. Technical Advisory Committee on Corrosion.

ACKNOWLEDGMENTS

To the engineers and chemists, and there were many who had no intimate concern with these studies but who, nevertheless, volunteered many helpful suggestions and much sound advice, the authors are indebted. Acknowledgment is due to Dr. J. C. Warner, head of the department of chemistry, Carnegie Institute of Technology, for his patience and counsel particularly in the early days, and to Dr. J. P. Fugassi for valuable assistance and suggestions. The generosity of G. D. Winans, engineer of steam distribution, The Detroit Edison Co., in loaning the experimental units, and of C. H. Fellows, head of the chemistry division, Research Department of the same company, in loaning an evolution carbonate determination apparatus, is also acknowledged.

APPENDIX

CONCERNING DETERMINATION OF CARBON DIOXIDE

In these studies, the determination of CO_2 involved the analysis of samples over an extremely wide range of concentrations. Because the accuracy of the methods commonly used to determine CO_2 have been for some time the source of considerable conjecture amongst analytical chemists, it seems entirely proper to record the experiences gained during these studies wherein several methods were used. Such is the purpose of this appendix.

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[8] Loc. Cit. Note 2.
[9] RI Report by A. A. Berk (U. S. Bureau of Mines, 3754, June, 1944).

Undissolved Gas

In the first series of tests (Heater No. 1), the steam supply contained 3 to 4 ppm of CO_2 and considerable air.[10] As a result, the CO_2 in the noncondensable gases vented from the unit consisted of only approximately 20 per cent CO_2. When the noncondensable gases were collected in a eudiometer tube over sulphuric acid (0.5 per cent solution) and subsequently analyzed in a Hempel apparatus, satisfactory balances were obtained. When, however, the steam was enriched so as to contain 40 ppm of CO_2—thus causing the noncondensable gases to contain about 75 per cent CO_2—this method did not give good balances. It is believed the inaccuracies resulted from solution of CO_2 in the acid.

Probable Accuracy of the Analytical Data

In the tests involving steams containing CO_2 of the order of 4 ppm, no data were accepted in which the CO_2 leaving the unit via the vent plus that dissolved in the condensate differed from that entering the unit by more than 20 per cent. For CO_2 values in the incoming steam of a higher order, data differing by more than 10 per cent were rejected. Better precision could not be hoped for because of the inherent limitations of the methods available for the determinations of small concentrations of dissolved CO_2. Since the CO_2 entering the experimental condensers was determined by a method entirely different from that used in measuring the outgoing CO_2, the possibility of accidental balances due to consistent errors is precluded.

By passing the vented gases (leaving the modified Hopkins condensers) through a magnesium perchlorate filled drying tube and absorbing the CO_2 in Ascarite, satisfactory balances were obtained for CO_2 concentrations within the range 2000 to over 20,000 ppm.

Dissolved CO_2

The *ASTM* evolution method was found satisfactory but time consuming on condensates containing up to about 1200 ppm of CO_2. On condensates not otherwise contaminated equally good results were obtained when an excess of standard $Ba\ (OH)_2$ was added to a 200 cc sample and the excess subsequently back-titrated with standard hydrochloric acid using phenolphthalein as indicator.

When alkaline materials were present, such as results from the entrainment of boiler water salines, it was found that titration of samples between pH 8.5 to 5.0, using a double indicator solution of o-cresolphthalein and methyl red, gave satisfactory CO_2 balances.[11] In retrospect, however, it is believed preferable to determine the bulk of the CO_2 content of steam by separation of the gas, using the Hopkins condenser modification B, and subsequent absorption of it in Ascarite. The relatively small amount of CO_2 in the condensate from the condenser can be determined by titration.

DISCUSSION[12]

L. F. COLLINS: In searching for a capable discusser of these two papers we were fortunate in finding that a few years ago some comparable research work of a laboratory nature was done by the Consolidated Gas and Electric Light and Power Co., Baltimore. Dr. Guernsey, who was responsible for the work, was kind enough to give both Mr. McKinney and myself a copy of the results, but as the manuscript received indicated that it was confidential information we did not feel at liberty to cite work to which we had privileged access. Since that time, however, it has been

[10] Because the Boiler feedwater, from which the steam was derived, is not deaerated.
[11] Determination of Carbon Dioxide in Water, by D. S. McKinney and A. M. Amorosi. (Industrial & Engineering Chemistry, Vol. 16, p. 315, May, 1944.)
[12] This discussion also covers Chapter No. 1265, p. 39.

decided that the results can be made public and therefore Dr. Guernsey will refer to them.

E. W. Guernsey, Baltimore, Md. (Written): The authors of these two papers deserve the thanks of all those concerned with corrosion in heating systems for the large amount of conscientious effort which has gone into the work described. They have made a substantial addition to the experimental data concerning the mechanism of entry of carbon dioxide into condensate and the means for keeping dissolved carbon dioxide at a minimum.

It may, or may not, be found generally feasible to vent a small part of the steam from heating equipment to reduce corrosion. In any case, the data presented have a very practical value in that they contribute to the fuller understanding of the factors which influence the solution of carbon dioxide and other gases in the condensate. Such an understanding should make it possible to take this factor into account in designing equipment. It will also often make possible the intelligent appraisal of the tendency toward corrosion to be expected with particular equipment or methods of operation.

The papers which have been presented afford additional assurance that there are no important unrecognized factors affecting the solution of gases in steam-heated equipment. We have, over a period of years, studied the tendency toward corrosion in various types of steam-heating equipment and have been able, in most instances, to rationalize the finding of high concentrations of carbon dioxide in condensate, and accelerated corrosion in particular systems, and in many cases to indicate the direction in which to seek improvement. This could be done by considering the method of operation of the equipment and the pattern of steam and condensate flow, and by applying the same principles employed in the two papers which have been presented. In addition, some laboratory studies were made under conditions permitting a better control of variables than was possible in the field. These results were described and given limited circulation in 1938, but were not published. That work and the studies which have been presented are in many respects supplementary. For that reason, it may be worth while to recall briefly some of our conclusions.

In the matter of venting, we sought first to establish the theoretical limitations of this practice. For a hypothetical case in which each element of steam is assumed to move forward, in contact with its own condensate, with the contained gases distributed in equilibrium between steam and condensate and with a final small portion of uncondensed steam withdrawn, it is of course possible to calculate from data on the solubility of carbon dioxide the fraction of contained carbon dioxide which would enter the condensate. The effectiveness of venting steam, calculated for such an ideal case, is shown in Fig. A.

Note that with steam condensing at atmospheric pressure the elimination of about 98 per cent of the carbon dioxide by venting one per cent of the steam is theoretically possible. Other calculations not shown were made for higher steam pressures. At 40 lb per square inch, gage, the theoretical elimination would be 95 per cent with one per cent purge.

Note further, however, that the effectiveness of venting is greatly reduced if the steam removed is not at the very end of the path of the steam flow in the apparatus. If as little as 0.1 per cent of the steam condenses *downstream* from the point of venting, the theoretical elimination with one per cent purge drops to 89 per cent, and with one per cent of such *downstream-condensation* the maximum elimination is only 50 per cent. This suggests that venting would lose its effectiveness unless obtained from a point just over the condensate level. This practice would be difficult except in connection with a type of trap which maintains a definite level of condensate.

It is, of course, not to be expected that carbon dioxide would, in practice, be eliminated by purging with the effectiveness calculated for the ideal case, at least not with equipment in which condensate and steam flow in the same direction. It was established experimentally, however, that the conclusions from the calculation were essen-

tially correct in that a high proportion of the carbon dioxide could be eliminated with a small fraction of steam *bled*. For this purpose a laboratory condenser such as is illustrated in Fig. B was used. This is simply a one-inch water-cooled tube B, terminating in a vertical ⅜-in. tube C, acting as a condensate receiver. Steam was supplied at A. Manual operation of valve D maintained the desired condensate level, as indicated on level indicator G, in the receiver C, and thus insured against any leaking of steam with the condensate. The vented steam was taken from the space

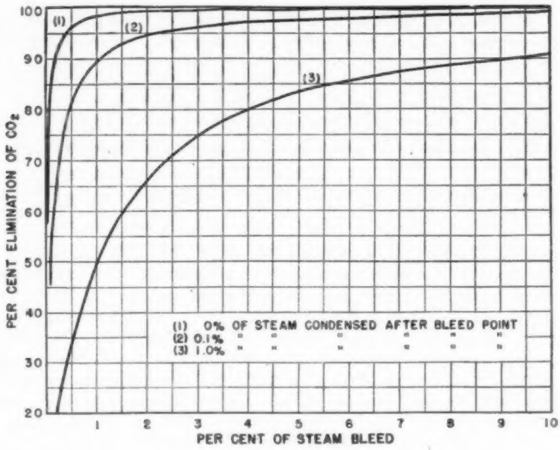

FIG. A. ELIMINATION OF CARBON DIOXIDE BY BLEEDING AT ATMOSPHERIC PRESSURE

above the condensate through the line E. Condensate was withdrawn for analysis through the cooling coil H and valve F. Typical results are shown in Table A.

Note that the actual elimination varies from 91 to 93 per cent, as compared to the calculated values for the ideal case of 96 to 99 per cent. This, then, establishes that, as predicted by the theory, a high elimination was possible in apparatus of this particular design. It did not indicate necessarily what it might be possible to do with equipment of commercial design. The observations presented by Mr. McKinney show, however, that with a commercial water heater of conventional design, a substantial elimination can be effected by properly arranged purging means.

TABLE A—ELIMINATION OF CARBON DIOXIDE BY VENTING

Pressure, lb/in.²	Per Cent Bled	Per Cent Elimination of CO_2	
		Calculated	Observed
40	1.2	96.0	90.6
40	2.6	98.1	91.2
40	3.8	98.7	92.4
8	2.1	98.8	93.1

The applicability of venting for preventing corrosion in many other types of steam heating equipment is not necessarily indicated, and, in fact, it is to be expected that venting will not be practicable in some cases. As one difficulty, the necessity for locating the vent at the end of the path was mentioned and this has also been stressed in the papers presented. This point is qualitatively shown by the results of some experiments with the apparatus of Fig. B modified by the removal of the insulation from the vertical condensate receiver, C, thus allowing it to act as an air condenser.

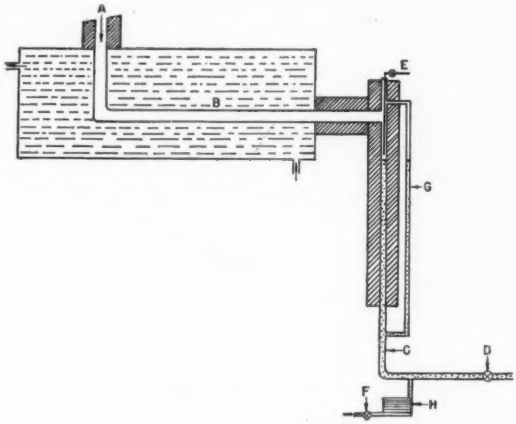

FIG. B. EXPERIMENTAL CONDENSER

denser. The condensate level was held at various distances below the vent point and the effect on the elimination of carbon dioxide noted. The results are shown in Table B.

The unknown but presumably small amount of *downstream-condensation* occurring in 19.5 in. of receiver tube reduced the elimination with 1.1 per cent purge to 49 per cent, as compared with 92 per cent when *downstream-condensation* was avoided. It is therefore apparent from these observations, as well as from the theoretical considerations mentioned previously, that the location of the vent is critical. It is a corollary that effective venting is difficult or impossible in systems in which the pattern of steam flow is complex, as in many radiator systems. Generally, however, the type of equipment in which corrosion difficulties most frequently occur is of a design most favorable for the application of venting.

TABLE B—EFFECT OF *Downstream-Condensation* ON ELIMINATION OF CARBON DIOXIDE BY BLEEDING

PRESSURE, PSI	PER CENT BLED	DISTANCE CONDENSATE LEVEL BELOW BLEEDING POINT-INCHES	PER CENT ELIMINATION OF CO_2
40	1.1	19.5	49
40	1.0	12.5	54
40	1.0	4.6	67
40	1.1	1.0	92

In much of the discussion of the effect of carbon dioxide in steam heating systems, it seems not to have been generally appreciated that under some circumstances it is possible to form condensate at certain points in a system containing a much higher proportion of carbon dioxide than is present in the original steam. The author refers to one such case—a one-pipe system operated for a long time without interruption. A very high concentration of carbon dioxide may occur in the radiator because of the stripping of the carbon dioxide from the condensate of the return line and its return to the radiator by the rising steam. Instances have been noted in which the nipples joining radiator sections were so severely corroded that it was necessary to replace them. This is not commonly experienced in one-pipe systems, because in most

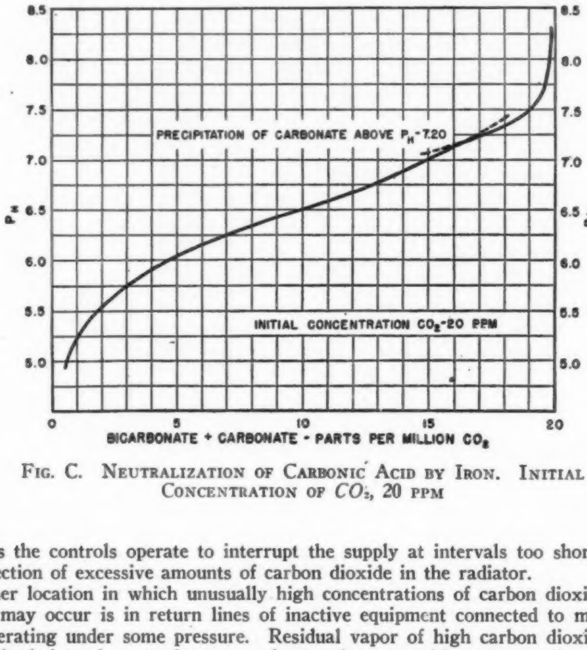

FIG. C. NEUTRALIZATION OF CARBONIC ACID BY IRON. INITIAL CONCENTRATION OF CO_2, 20 PPM

instances the controls operate to interrupt the supply at intervals too short to allow the collection of excessive amounts of carbon dioxide in the radiator.

Another location in which unusually high concentrations of carbon dioxide in condensate may occur is in return lines of inactive equipment connected to main return lines operating under some pressure. Residual vapor of high carbon dioxide content flowing back into these *pockets* may give condensates with concentrations of carbon dioxide as high as several hundred parts per million. The possibility of such high concentration of carbon dioxide is of particular concern because it is believed that there may be more than a proportional increase in corrosion with increasing carbon dioxide. The reason is that when iron is attacked by high concentrations of carbon dioxide the buffering action of iron corrosion product is not fully effective, because of the precipitation of iron carbonate. It is estimated that when carbon dioxide solution reacts with iron in the absence of oxygen at about 75-80 F, with concentrations of either 20 or 100 ppm, a pH of 6 is reached when about 24 per cent of the carbon dioxide has reacted with iron. On the other hand, with 1000 ppm of carbon dioxide in solution, it is estimated that a pH of 6 will not be reached until nearly 75 per cent of the carbon dioxide has reacted with iron. The data are not available for similar calculations for the higher temperatures corresponding to condensate in the return lines. It is probable that a precipitation of ferrous carbonate would likewise occur

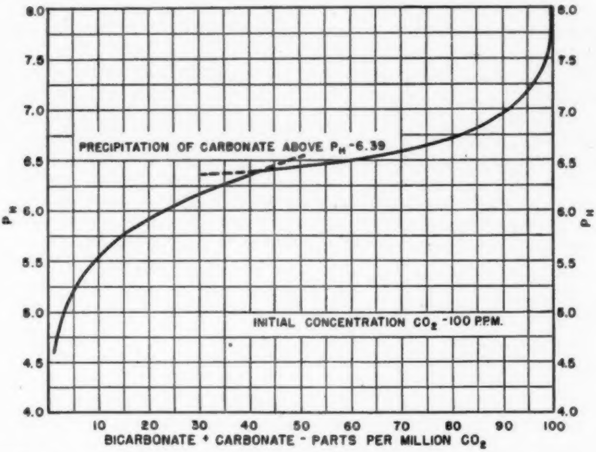

FIG. D. NEUTRALIZATION OF CARBONIC ACID BY IRON. INITIAL CONCENTRATION OF CO_2, 100 PPM

with higher concentration of carbon dioxide, although the quantitative relationships would be changed. The calculated curves for change in pH on continued action of carbon dioxide solution of the three different concentrations at about 75 F are shown in Figs. C, D and E. The calculations are based on available information, including data on the dissociation of carbonic acid and the solubility of ferrous carbonate.

There are one or two points in the data offered in the papers presented for which alternate interpretations might be considered. Thus it is not clear that, in the pres-

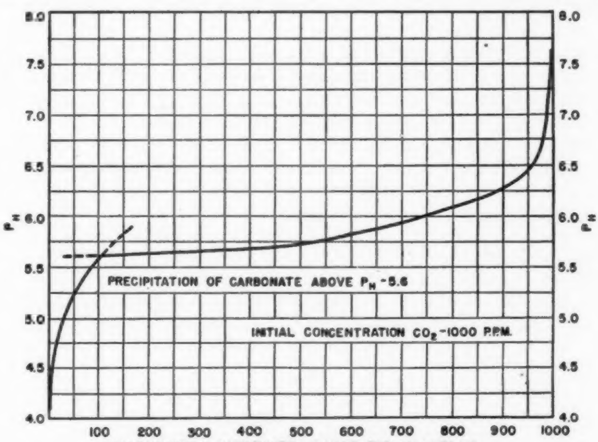

FIG. E. NEUTRALIZATION OF CARBONIC ACID BY IRON. INITIAL CONCENTRATION OF CO_2, 1000 PPM

ence of steam, sufficient undercooling would occur to significantly affect the entry of carbon dioxide into the condensate in the experimental water heaters, as suggested by the authors. In any case the concentration of carbon dioxide in the condensate is from five to fifty times that which would be in equilibrium with the vented steam. This must mean that much of the condensate has traveled through regions containing a higher partial pressure of carbon dioxide than is present at the point of purging. This factor is recognized by the authors and is perhaps alone sufficient to account for the variations noted without assuming any effects from *undercooling*.

The loss of CO_2 in passing through the meter noted in the Appendix of Mr. Collins' paper may be primarily a degassing effect due to the sweeping action of air. It does not seem probable that iron corrosion products would substantially affect the solubility of CO_2, except insofar as it is combined as iron bicarbonate.

FERDINAND JEHLE, Indianapolis, Ind.: I did not notice just how Dr. Guernsey controlled the point where he vented. It undoubtedly was on the second slide, but I think that I missed it.

P. H. WEITZEL, Dayton, Ohio: What is the generally accepted source of the gas? Is it dirty water? Is there any other way of solving the problem besides attempting to vent in some complicated manner?

DR. GUERNSEY: In the experimental apparatus (Fig. B) we controlled the level of the concentrate, observing it in a gage glass and then controlled it manually during the experiment simply by manipulating the valve F. The vertical tube was the line through which we vented steam. Then by manipulating the valve to maintain the level of the condensate a very short distance under that vent point, we could be certain that we were venting from practically the end of the path of travel of the steam.

Of course, likewise we could open the valve in such a way that the condensate level could be carried at any desired lower point.

As I explained for that particular observation, we removed insulation from this tube to permit a small amount of condensation beyond this point. We do not know how much as we did not measure the amount. But we assumed that as compared with the cooling by the condensation of the water-cooled tube some condensation occurred beyond this point. Yet it was found that about 19 in. of this three-eighths inch tube was sufficient to reduce the venting from about 92 per cent to 49 per cent.

MR. JEHLE: What is part H in Fig. B?

DR. GUERNSEY: That merely indicates the means we used for sampling this condensate. We were operating at pressures above atmospheric in all the experiments. As we wanted to take the condensate out and analyze it, we used a cooling coil H through which we withdrew the condensate into our sampling bottles.

MR. COLLINS: To answer the question as to the source of CO_2 in the steam: The principal source of CO_2 in steam is the carbonate salts (usually calcium or magnesium bicarbonate) that are present in the raw water fed to the boiler. Under the conditions of operation these salts are decomposed with the liberation of CO_2.

D. M. HUMMEL, New Haven, Conn.: I would like to ask if CO_2 corrosion has any distinctive symptoms. One sees two kinds, for instance, grooving on the bottom of the pipe, and holes on the top.

MR. COLLINS: In general the pattern of oxygen corrosion is a pitted surface over which there is an accumulation of insoluble deposits. Carbon dioxide corrosion is typified by clean, evenly thinned surfaces. In partially filled lines, there is usually a grooving along the bottom.

No. 1267

SUMMER WEATHER DATA AND SOL-AIR TEMPERATURE—STUDY OF DATA FOR NEW YORK CITY

By C. O. Mackey * and E. B. Watson,** Ithaca, N. Y.

THIS IS a discussion of a method of analysis of available summer weather data in order to obtain information of some value to engineers interested in the effect of solar heat upon cooling load and in other related problems. The intensity of the total radiation received from sun and sky on a horizontal surface is continuously recorded at several stations in the United States, maintained by the U. S. Weather Bureau and also at a number of cooperating stations. Hourly readings of the temperature of the outdoor air are also recorded. The intensity of direct solar radiation received in a plane perpendicular to the sun's rays is measured at four of these stations. Monthly summaries of these data appear in the Monthly Weather Review.

The sol-air temperature seems to be the most logical combination of these readings in analyzing problems in heat transfer. As explained in an earlier paper,† the *sol-air temperature* is the temperature of the outdoor air which, in contact with the shaded surface of any building material that does not directly transmit solar radiation, would give the same rate of heat transfer and the same temperature distribution through that material as exists with the actual outdoor air temperature and solar radiation incident upon the sunlit surface.

For either steady or unsteady flow of heat, the instantaneous rate of heat entry into the outside surface of a sunlit building material, which does not directly transmit solar radiation, is:

$$\left[\frac{q}{A}\right]_L = bI + h\,(t_a - t_L) \text{ Btu/hr ft}^2,$$

where

b = absorptivity of the surface for solar radiation;
I = intensity of incident solar radiation, Btu/hr ft^2;
h = film coefficient of heat transfer between outdoor air and surface of the material Btu/hr ft^{2}F;
t_a = temperature of outdoor air, F;
t_L = temperature of surface of material, F.

Note that:

$$bI + h(t_a - t_L) = h\left[\frac{bI}{h} + t_a - t_L\right]$$

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** Assistant Professor of Engineering Materials, Cornell University.
† Summer Comfort Factors as Influenced by the Thermal Properties of Building Materials, by C. O. Mackey and L. T. Wright, Jr. (A.S.H.V.E. Transactions, Vol. 49, 1943, p. 148.)

Presented at the 51st Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, Mass., January, 1945.

Let the sol-air temperature be defined as:

$$t_e = t_a + \frac{bI}{h}$$

Then, the instantaneous rate of heat entry into the material at the outside surface may also be expressed as

$$\left[\frac{q}{A}\right]_L = h(t_e - t_L)$$

The sol-air temperature concept is new but extremely useful, because the effects of air temperature and solar radiation upon the rate of heat transfer and the temperature distribution through the sunlit material are combined. In other words, an increment in one degree in the modulus, $\frac{bI}{h}$, which has the dimensions of temperature, has precisely the same effect upon the temperature

TABLE 1—OUTDOOR AIR TEMPERATURE DATA—NEW YORK CITY

(Maximum hourly outdoor air temperature in the shade, and hourly outdoor air temperature equalled or exceeded on no more than 5 per cent of the days during a given month in the period from 1932 through 1941)

TIME OF DAY (E.S.T.)	TEMPERATURE OF THE OUTDOOR AIR IN THE SHADE, DEG F							
	MAXIMUM				EQUALLED OR EXCEEDED ONLY 5 PER CENT OF THE TOTAL HOURS, 1932–1941			
	June	July	August	September	June	July	August	September
1 AM	80	84	85	79	76	78	77	73
2	78	82	84	78	75	77	76	72
3	77	81	83	77	74	77	75	72
4	77	81	82	77	73	76	75	71
5	77	80	81	75	74	76	74	71
6	81	82	82	75	75	76	75	71
7	84	86	84	78	7[illegible]	80	78	72
8	86	90	86	80	80	82	80	74
9	90	92	91	83	82	86	83	77
10	91	95	93	86	84	88	86	80
11	94	98	95	89	87	90	88	83
12	96	100	96	90	88	92	90	85
1 PM	97	102	98	93	90	93	91	87
2	98	104	100	96	91	94	91	87
3	99	105	99	96	90	94	91	88
4	100	106	96	92	89	94	91	86
5	97	105	97	90	89	93	90	85
6	94	103	96	87	87	90	88	82
7	92	94	94	86	84	88	85	79
8	88	93	93	84	82	85	83	77
9	86	92	91	83	80	83	81	76
10	85	88	89	82	79	82	80	75
11	83	85	87	80	78	81	78	74
12	81	84	86	79	76	79	77	73
24-hour average of above	88.0	92.1	90.3	84.0	81.7	84.8	82.6	77.9

TABLE 2—INTENSITY OF TOTAL SOLAR AND SKY RADIATION ON A HORIZONTAL SURFACE —NEW YORK CITY

(Maximum hourly intensity, and hourly intensity equalled or exceeded on no more than 5 per cent of the days during a given month in the period from 1932 through 1941; all times are Eastern Standard)

Time of Day	Intensity of Total Solar and Sky Radiation on Horizontal Plane, Btu/hr ft²							
	Maximum				Equalled or Exceeded Only 5 Per Cent of the Total Hours, 1932–1941			
	June	July	August	September	June	July	August	September
6 AM	45	51	28	...	30	28	15	...
7	127	127	84	52	91	80	59	31
8	206	160	146	154	153	144	126	87
9	279	216	211	183	205	196	180	151
10	318	268	261	224	249	236	210	197
11	347	295	288	252	287	260	244	224
12	362	305	301	270	302	278	266	241
1 PM	360	300	305	279	287	278	264	239
2	343	289	292	268	260	267	241	221
3	326	258	256	226	280	225	216	186
4	258	215	206	183	215	189	171	150
5	187	166	142	118	163	137	123	91
6	112	108	84	66	85	82	65	37
7	38	44	31	...	35	31	19	...

of the *inside* surface, and the rate of heat transfer at that surface contributing to the cooling load, as would an increment of one degree in the temperature of the outdoor air.

A more complete discussion of sol-air temperature, with numerical examples, is given in Appendix—*A* for materials which do not directly transmit solar radiation, and in Appendix—*E* for glass

Hourly readings of the dry-bulb temperature of the outdoor air and the total solar and sky radiation incident upon a *horizontal* surface for each hour of each day during the months of June, July, August, and September for the ten-year period from 1932 through 1941 were obtained from data reported by the New York Meteorological Observatory located in Central Park at a north latitude of 40°46′, a west longitude of 73°58′, and at an elevation of 180 ft above sea level.‡ Considerable atmospheric contamination reduces the intensity of solar radiation received at this station, but Dr. I. F. Hand of the U. S. Weather Bureau believes that *values representative of average large city conditions* are obtained.

Table 1 gives temperature data; the maximum temperature at each hour during June, for example, is the maximum of 300 recorded temperatures; the temperature listed as equalled or exceeded by that at only 5 per cent of the total hours would be that temperature equalled or exceeded in June, for example, by 15 of the 300 readings for that hour.

Table 2 gives solar radiation data; the maximum intensity of total solar and

‡ Tables presented here are taken from a thesis by E. B. Watson for the degree of Master of Science in Engineering at Cornell University.

sky radiation received on a horizontal surface is listed for each hour during each month of the ten-year period. Also the hourly intensity equalled or exceeded on no more than 5 per cent of the days at a given hour during each month of the ten-year period is given.

Table 3 gives the maximum sol-air temperature for the months of June, July, August and September for a horizontal surface with three solar absorptivities 1.0, 0.7 and 0.4. All sol-air temperatures are based upon Equation 1 of this report (Appendix—*A*). Careful observers may note what appears to be a discrepancy in Tables 1, 2 and 3. For example, *if* the maximum air temperature and the maximum solar radiation had occurred *simultaneously* at 12 noon in June, the maximum sol-air temperature for the horizontal surface with solar absorptivity, *b*, of 1.0 would have been $\left\{96+\frac{362}{4}\right\}$ or 186.5 F; actually, the maximum sol-air temperature for this hour and month was observed to be only 162 F. The reason for this is the observed fact that *the maximum air temperatures and the maximum solar radiation do not, in general, occur simultaneously.* It is not correct, therefore, to combine maximum

TABLE 3—MAXIMUM SOL-AIR TEMPERATURE FOR HORIZONTAL SURFACE IN NEW YORK CITY

(*Maximum values for 10-year period, 1932–1941*)

TIME OF DAY (E.S.T.) (SOLAR ABSORPTIVITY)	SOL-AIR TEMPERATURE, DEG F											
	June			July			August			September		
	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)
1 AM...	80	80	80	84	84	84	85	85	85	79	79	79
2 ...	78	78	78	82	82	82	84	84	84	78	78	78
3 ...	77	77	77	81	81	81	83	83	83	77	77	77
4 ...	77	77	77	80	80	80	82	82	82	77	77	77
5 ...	77	77	77	82	82	82	81	81	81	75	75	75
6 ...	86	85	85	86	85	83	86	85	84	75	75	75
7 ...	102	95	90	104	98	92	98	94	90	89	85	81
8 ...	125	111	96	118	108	100	113	103	95	107	95	86
9 ...	141	123	106	139	124	108	130	117	105	119	106	95
10 ...	149	127	111	148	132	116	145	128	111	134	118	103
11 ...	155	135	115	157	139	122	150	132	116	145	126	108
12 ...	162	140	118	162	143	125	160	141	121	150	131	112
1 PM...	163	145	122	170	146	126	161	141	121	151	132	114
2 ...	164	142	122	163	145	127	161	141	121	150	137	115
3 ...	155	137	120	169	150	131	153	135	118	141	125	111
4 ...	143	129	117	144	133	121	140	127	113	128	117	106
5 ...	131	121	110	131	123	116	125	115	106	111	104	98
6 ...	115	108	102	113	110	107	109	104	99	95	93	90
7 ...	100	98	95	100	98	96	94	96	95	86	86	86
8 ...	88	88	88	93	93	93	93	93	93	84	84	84
9 ...	86	86	86	92	92	92	91	91	91	83	83	83
10 ...	85	85	85	88	88	88	89	89	89	82	82	82
11 ...	83	83	83	85	85	85	87	87	87	80	80	80
12 ...	81	81	81	84	84	84	86	86	86	79	79	79
24-hour average of above.	112.6	104.5	96.7	114.8	107.7	100.9	111.9	105.0	98.2	103.1	96.8	90.6

values of the two effects and the need for some concept like the sol-air temperature becomes apparent. Another apparent discrepancy may be found if one assumes that the difference between the sol-air temperatures for $b = 1$ and $b = 0.7$ should be the same as the difference for $b = 0.7$ and $b = 0.4$; this need not be the case, however, for the temperature of the air becomes more important when the solar absorptivity decreases. In other words, the maximum sol-air temperature for $b = 1$ might occur on the day when the solar radiation is a maximum, while for $b = 0.4$ the sol-air temperature may be a maximum on the day when the temperature of the air is a maximum. This latter effect is small and for most practical purposes the difference between the sol-air temperature for any solar absorptivity and the sol-air temperature for zero absorptivity (air temperature in the shade) may be assumed to be directly proportional to that solar absorptivity.

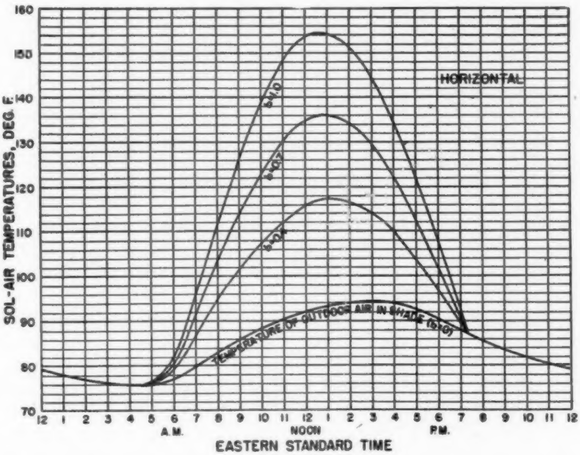

FIG. 1. DESIGN TEMPERATURES FOR HORIZONTAL SURFACE IN NEW YORK, N. Y.

Since the 24-hour average of the sol-air temperature is greater for July than for any other month, the sol-air temperature at each hour in July which is equalled or exceeded at that hour only 16 times in 310 observations in July (ten-year period) has been chosen as the design sol-air temperature of the outdoor air. These temperatures are plotted in Fig. 1 for a horizontal surface and the result is a design curve for New York City which is recommended for use in calculations of heat transfer through horizontal surfaces in that locality. Similar curves must be found for other localities before any general design curves may be drawn.

The design curves include the air temperature in the shade (solar absorptivity of zero) and three curves of sol-air temperatures for solar absorptivities of 1.0, 0.7 and 0.4. Although data on solar absorptivity are not very complete, it is recommended that a solar absorptivity of 1.0 be used for very dark colors, 0.7 for medium colors and 0.4 for very light colors; the air tempera-

ture curve should be used for all surfaces completely shaded from direct solar and diffuse sky radiation.

Vertical Surfaces

It would be desirable to have the sol-air temperature of the outdoor air for vertical surfaces of various orientations in order that the flow of heat through

Table 4—Sol-Air Temperature for Horizontal Surface in New York City Equalled or Exceeded at Any Hour Only 5 Per Cent of the Total Hours in a Ten-Year Period, 1932–1941

Time of Day (E.S.T.) (Solar Absorptivity	Sol-Air Temperature, deg F											
	June			July			August			September		
	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)
1 AM...	76	76	76	78	78	78	77	77	77	73	73	73
2 ...	75	75	75	77	77	77	76	76	76	72	72	72
3 ...	74	74	74	77	77	77	75	75	75	72	72	72
4 ...	73	73	73	76	76	76	75	75	75	71	71	71
5 ...	74	74	74	76	76	76	74	74	74	71	71	71
6 ...	79	78	76	81	79	78	78	77	77	71	71	71
7 ...	93	88	83	96	92	87	89	85	81	78	77	75
8 ...	108	99	91	110	102	93	106	98	90	91	84	82
9 ...	123	111	99	127	114	101	122	111	97	106	97	88
10 ...	137	121	105	137	122	108	131	117	103	121	108	94
11 ...	143	127	109	148	130	112	141	124	108	131	116	103
12 ...	149	131	112	155	135	116	148	129	113	136	121	106
1 PM...	148	130	116	154	136	118	150	132	114	141	124	108
2 ...	150	132	113	152	134	116	146	130	113	136	121	106
3 ...	141	125	110	144	129	113	140	125	110	128	115	102
4 ...	132	119	106	136	123	110	129	118	106	118	107	97
5 ...	121	111	101	121	112	104	116	108	100	101	95	91
6 ...	104	99	94	106	101	96	101	97	91	87	85	83
7 ...	92	89	87	93	91	90	88	87	86	79	79	79
8 ...	82	82	82	85	85	85	83	83	83	77	77	77
9 ...	80	80	80	83	83	83	81	81	81	76	76	76
10 ...	79	79	79	82	82	82	80	80	80	75	75	75
11 ...	78	78	78	81	81	81	78	78	78	74	74	74
12 ...	76	76	76	79	79	79	77	77	77	73	73	73
24-hour average of above.	103.6	97.0	90.3	106.4	99.8	93.2	103.0	96.4	90.2	94.1	88.9	84.2

such surfaces might be found. The Weather Bureau has made very few observations of the intensity of solar radiation received on vertical surfaces. If it were not for the fact that part of the total observed radiation received upon the horizontal surface is diffuse or sky radiation, it would be relatively simple to calculate the intensity on a vertical surface from the observations on the horizontal surface. (For direct or beamed solar radiation, the ratio of the intensity on a vertical surface to the intensity on a horizontal surface may be found as explained in Appendix—*B*).

As solar radiation passes through the earth's atmosphere, part is turned aside from the direct beam and scattered in practically all directions with no

appreciable change in wave length. A considerable portion of the total radiation received on a horizontal surface is in the form of this diffuse radiation from the sky. On cloudy days all radiation is diffuse, while on clear days the ratio of diffuse to direct radiation varies with the solar altitude and with the amount of dust, water vapor and other material in the atmosphere. This ratio is large for low solar altitudes, cloudy days and large amounts of smoke

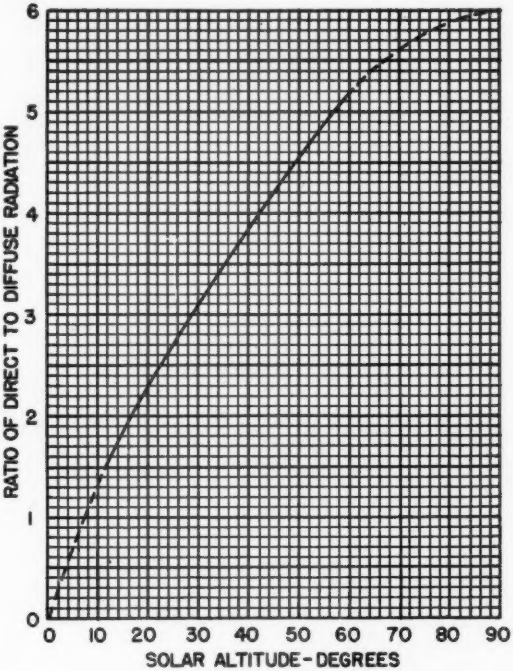

FIG. 2. RATIO OF DIRECT TO DIFFUSE RADIATION ON HORIZONTAL SURFACES; SUMMER DATA; EASTERN STATES

and dust in the atmosphere. On clear days, in cities near the times of sunrise and sunset, the diffuse radiation generally amounts to more than one-half of the total received on a horizontal surface, while on high mountains the diffuse radiation is almost negligible except with very low sun.

From a large number of measurements of diffuse radiation, Dr. I. F. Hand gives the following table [1] for the ratio of direct solar radiation on a horizontal surface to diffuse sky radiation on that surface during cloudless days in Washington, D. C. as indicated in the tabulation.

[1] Monthly Weather Review, 1937, page 437.

Solar altitude, H	60°	30°	11.3°
Ratio of direct to diffuse radiation, summer	5.2	3.1	1.5
Ratio of direct to diffuse radiation, winter	8.1	5.0	1.7

The summer results are shown plotted in Fig. 2, in the form of the ratio of direct to diffuse radiation received on a horizontal surface as a function of solar altitude. The resulting curve is typical for eastern states.

The point of greatest uncertainty in the extension of data obtained from radiation incident upon horizontal surfaces is the question of how the diffuse sky radiation on a vertical surface compares with the diffuse sky radiation on a horizontal surface. What fraction of this diffuse sky radiation on a horizontal surface is incident upon vertical surfaces of different orientations at different times of the day? Only more complete radiation data would answer this question. In this report, for simplicity, it has been assumed that the diffuse sky radiation incident on a vertical surface of any orientation is one-half of that incident upon a horizontal surface at the same time of day.

TABLE 5—SOL-AIR TEMPERATURES FOR VERTICAL SURFACES IN NEW YORK CITY EQUALLED OR EXCEEDED AT ANY HOUR IN JULY ONLY 5 PER CENT OF THE TOTAL HOURS IN A TEN-YEAR PERIOD, 1932–1941

(*Calculated values based upon assumptions stated in the body of the paper.*)

TIME OF DAY (E.S.T.) (SOLAR ABSORPTIVITY)	SOL-AIR TEMPERATURE, DEG F											
	SURFACE FACING											
	North			East			South			West		
	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)
1 AM...	78	78	78	78	78	78	78	78	78	78	78	78
2 ...	77	77	77	77	77	77	77	77	77	77	77	77
3 ...	77	77	77	77	77	77	77	77	77	77	77	77
4 ...	76	76	76	76	76	76	76	76	76	76	76	76
5 ...	76	76	76	76	76	76	76	76	76	76	76	76
6 ...	80	79	77	89	85	81	77	77	76	77	77	76
7 ...	85	83	82	106	98	91	82	82	81	82	82	81
8 ...	85	84	83	114	105	95	86	85	83	85	84	83
9 ...	90	89	88	120	109	99	97	94	90	90	89	88
10 ...	92	91	90	114	106	98	104	99	94	92	91	90
11 ...	94	93	92	106	101	97	111	105	98	94	93	92
12 ...	97	95	93	97	95	94	115	108	101	98	96	94
1 PM...	98	96	95	98	96	95	117	110	103	108	104	99
2 ...	99	97	96	99	97	96	112	107	101	126	116	107
3 ...	99	97	96	99	97	96	102	99	97	136	123	111
4 ...	99	97	96	99	97	96	99	98	96	145	130	114
5 ...	103	100	97	97	96	95	97	96	95	140	126	112
6 ...	106	101	96	93	92	91	93	92	91	134	120	107
7 ...	102	98	94	90	89	89	90	89	89	115	107	99
8 ...	85	85	85	85	85	85	85	85	85	85	85	85
9 ...	83	83	83	83	83	83	83	83	83	83	83	83
10 ...	82	82	82	82	82	82	82	82	82	82	82	82
11 ...	81	81	81	81	81	81	81	81	81	81	81	81
12 ...	79	79	79	79	79	79	79	79	79	79	79	79
24-hour average of above.	88.5	87.3	86.2	92.3	89.9	87.8	90.7	89.0	87.1	96.5	93.0	89.5

Based upon these assumptions, the sol-air temperatures were calculated for the month of July and for vertical surfaces of various orientations and are shown in Table 5. These are the sol-air temperatures that were equalled or exceeded on no more than 5 per cent of the days during that month in the ten-year period from 1932 to 1941. These values are not as reliable as those

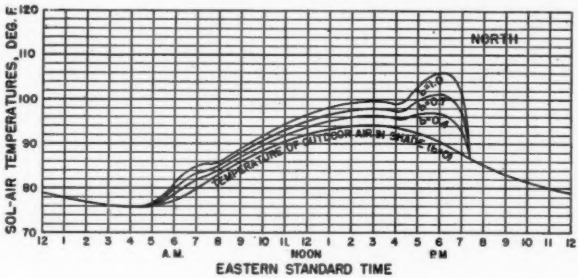

FIG. 3. DESIGN TEMPERATURES FOR SURFACE FACING NORTH IN NEW YORK, N. Y.

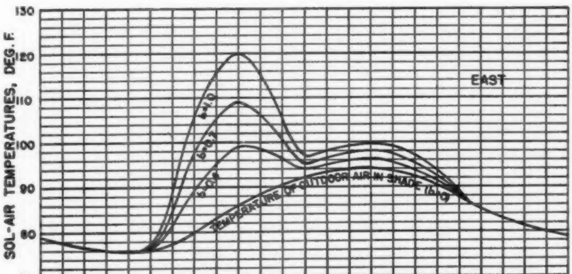

FIG. 4. DESIGN TEMPERATURES FOR SURFACE FACING EAST IN NEW YORK, N. Y.

for the horizontal surface, because they are based upon the assumed breakdown of total radiation into direct and diffuse and a further assumed ratio of diffuse radiation on the vertical surfaces to diffuse radiation on a horizontal surface. However, the curve of Fig. 2 is believed to be fairly reliable, and the effect of the assumption concerning the diffuse radiation incident upon the vertical surface is relatively unimportant in the final result as is shown in Appendix—*C*.

The recommended design curves for sol-air temperature for New York City are shown for vertical surfaces with different values of solar absorptivity facing north (Fig. 3), facing east (Fig. 4), facing south (Fig. 5) and facing west (Fig. 6).

EQUATIONS FOR SOL-AIR TEMPERATURE

For the few who may be interested, equations for the recommended design sol-air temperatures for New York City are presented in the form of trigo-

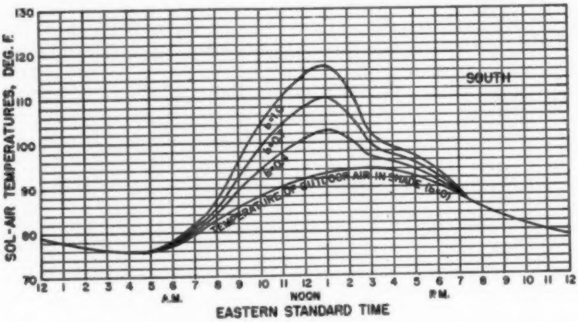

FIG. 5. DESIGN TEMPERATURES FOR SURFACE FACING SOUTH IN NEW YORK, N. Y.

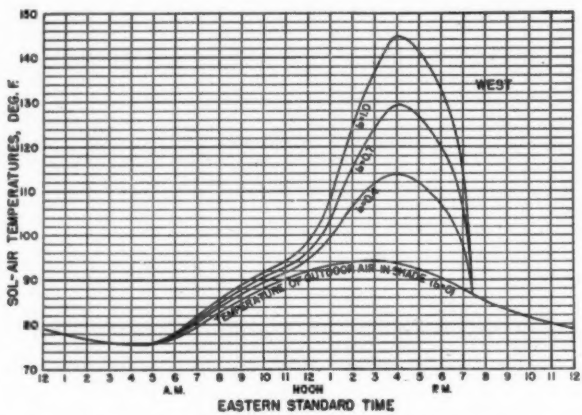

FIG. 6. DESIGN TEMPERATURES FOR SURFACE FACING WEST IN NEW YORK, N. Y.

nometric series. In all of these equations, θ represents the time in hours after noon (E.S.T.).

Design temperature of outdoor air in shade:

$$t_a = 84.8 + 9.0 \cos (15\theta - 42) + 1.1 \cos (30\theta - 28) + 0.7 \cos (45\theta - 228) + 0.3 \cos (60\theta - 225)$$

Design sol-air temperature of outdoor air for horizontal surfaces:

Solar absorptivity of 1.0,

$$t_e = 106.4 + 39.7 \cos(15\theta - 15) + 11.5 \cos(30\theta - 17) + 2.0 \cos(45\theta - 210) + 1.4 \cos(60\theta - 217)$$

Design sol-air temperature of outdoor air for vertical surfaces facing *North:*

Solar absorptivity of 1.0,

$$t_e = 88.5 + 12.9 \cos(15\theta - 44) + 1.8 \cos(30\theta - 165) + 2.6 \cos(45\theta - 253) + 2.1 \cos(60\theta - 12)$$

Design sol-air temperature for vertical surfaces facing *East:*

Solar absorptivity of 1.0,

$$t_e = 92.3 + 15.5 \cos(15\theta - 356) + 6.9 \cos(30\theta - 262) + 8.3 \cos(45\theta - 209) + 2.6 \cos(60\theta - 154)$$

Design sol-air temperature for vertical surfaces facing *South:*

Solar absorptivity of 1.0,

$$t_e = 90.7 + 17.6 \cos(15\theta - 22) + 6.3 \cos(30\theta - 5) + 2.3 \cos(45\theta - 284) + 0.9 \cos(60\theta - 5)$$

Design sol-air temperature for vertical surfaces facing *West:*

Solar absorptivity of 1.0,

$$t_e = 96.5 + 28.1 \cos(15\theta - 54) + 12.6 \cos(30\theta - 125) + 8.0 \cos(45\theta - 203) + 2.2 \cos(60\theta - 281)$$

SOL-AIR TEMPERATURE FOR GLASS

The sol-air temperatures given in this paper are to be used only for estimating the rate of heat flow from the indoor surfaces of materials which do not directly transmit incident solar radiation. Part of the total radiation incident upon glass is directly transmitted, part is reflected and part is absorbed. If the sol-air temperature concept is to be used for predicting the heat flow through glass, this temperature must be increased to allow for the direct transmission of some of the incident radiation. Exact application requires a knowledge of the transmissivity and absorptivity of glass as a function of the angle of incidence of the sun's rays. This point is discussed further in Appendix—*E*. The simplest procedure in obtaining design information concerning the contribution to the cooling load due to heat transfer through glass would probably be to multiply the values of total radiation received on a horizontal surface, as given in Table 2, by an appropriate factor at each hour; this factor would include the combined effects of glass orientation and the effect of angle of incidence upon absorptivity and transmissivity.

APPENDIX

METHODS AND EQUATIONS

A. Sol-Air Temperature

For the surface of a material which does not directly transmit solar radiation, the sol-air temperature may be found from observed values of the air temperature and the total solar and sky radiation incident upon that surface as follows:

$$t_e = t_a + \frac{bI}{h} \quad \text{. . . . . (1)}$$

where

t_e = the sol-air temperature, F;
t_a = the dry-bulb temperature of the outdoor air in the shade, F;
I = the intensity of total solar and sky radiation incident upon the surface, Btu/hr ft²;
b = the absorptivity of the surface for the incident total radiation;
h = the film coefficient of heat transfer between surface and air, Btu/hr ft² F. (Actually, this coefficient depends upon wind velocity, but it was assumed to be a constant and equal to 4 in this study.)

Example: Observed dry-bulb temperature of outdoor air in the shade is $t_a = 78$ F; observed total radiation incident upon *horizontal* surface is $I = 200$ Btu/hr ft². For a surface absorptivity of 0.7, the sol-air temperature for the *horizontal* surface is:

$$t_e = 78 + \frac{0.7\ (200)}{4} = 113\ \mathrm{F}$$

In other words, air at a dry-bulb temperature of 113 F in contact with the completely *shaded* surface would give the same temperature distribution and rate of heat flow through the building material as exists with an air temperature of 78 F, incident total radiation of 200 Btu/hr ft² and a surface absorptivity of 0.7. This concept permits combining the effects of air temperature and solar radiation.

B. Trigonometric Relations

The intensity of *direct* solar radiation on a horizontal surface is substantially equal to the product of the intensity of direct solar radiation received on a plane perpendicular to the sun's rays and the cosine of the angle between the sun's rays and the normal to the horizontal, or

$$I_H = K_H I_o \qquad (2)$$

where

I_H = the intensity of *direct* solar radiation received on a horizontal surface, Btu/hr ft²;
I_o = the intensity of *direct* solar radiation received on a plane perpendicular to the sun's rays, Btu/hr ft².

$$K_H = \cos (90 - H) \qquad (3)$$

or

$$K_H = \sin D \sin L + \cos D \cos L \cos (360 - \theta) \qquad (4)$$

where

H = the solar altitude, degrees;
D = the declination of the sun, degrees;
L = the latitude of the surface, degrees;
θ = the hour angle of the sun measured west from the zenith, degrees.

The intensity of *direct* solar radiation on a vertical surface is:

$$I_v = K_v I_o \qquad (5)$$

where

I_v = the intensity of *direct* solar radiation received on a vertical surface, Btu/hr ft²;

$$K_v = \cos H \cos B \qquad (6)$$

and

$$B = A_v - 90 - A_s \qquad (7)$$

where

A_v = the azimuth of the trace of the vertical surface, measured west from south, degrees;
A_s = the azimuth of the sun measured west from south, degrees, (If $\cos B$ is zero or negative, the vertical surface is receiving no *direct* solar radiation.)

The azimuth of the sun may be found from the following relation:

$$\sin (A_s - 180) = \frac{\cos D \sin (360 - \theta)}{\cos H} \qquad (8)$$

TABLE A—SOLAR ALTITUDES AT DIFFERENT TIMES FOR NEW YORK CITY IN MID-JULY

TIME (E.S.T.)	SOLAR ALTITUDE NORTH LATITUDE OF 40° 46′ WEST LONGITUDE OF 73° 58′ AND SOLAR DECLINATION OF 21° 36′
4:36 AM	0 (sunrise)
5	4° 2′
6	14° 40′
7	25° 47′
8	37° 7′
9	48° 21′
10	58° 56′
11	67° 29′
12 noon	70° 49′
1 PM	66° 31′
2	57° 33′
3	46° 50′
4	35° 33′
5	24° 14′
6	13° 10′
7	2° 38′
7:16	0 (sunset)

If the *direct* solar radiation received on a horizontal surface is known, the *direct* solar radiation that is then incident upon a vertical surface of any orientation may be computed.

Tables A and B used in the calculations are offered here because they may be of some interest in similar calculations. Table A gives the solar altitudes at different times for a north latitude of 40°46′ and a west longitude of 73°58′ (New York) and a solar declination of 21°36′ (mid-July).

TABLE B—RATIO OF DIRECT SOLAR RADIATION ON VERTICAL TO RADIATION ON HORIZONTAL SURFACE FOR NEW YORK CITY IN MID-JULY

TIME (E.S.T.)	RATIO: DIRECT SOLAR RADIATION ON VERTICAL SURFACE TO DIRECT SOLAR RADIATION ON HORIZONTAL SURFACE							
	VERTICAL SURFACE FACING							
	N	NE	E	SE	S	SW	W	NW
5 AM	6.044	13.335	12.814	4.787	...	...	...	...
6	1.059	3.346	3.673	1.848	...	...	...	...
7	0.254	1.633	2.055	1.273	...	...	...	...
8	...	0.894	1.321	0.973	0.056	...	...	...
9	...	0.460	0.864	0.762	0.213	...	...	...
10	...	0.167	0.526	0.580	0.295	...	...	...
11	...	...	0.243	0.481	0.335	0.065	...	...
12 noon	...	...	...	0.238	0.349	0.259	0.018	...
1 PM	...	...	...	0.003	0.333	0.433	0.280	...
2	...	...	...	...	0.287	0.604	0.568	0.205
3	...	...	...	...	0.196	0.788	0.918	0.510
4	...	...	...	...	0.028	1.009	1.399	0.970
5	...	...	...	...	...	1.326	2.198	1.783
6	...	...	...	...	...	1.987	4.083	3.787
7	...	...	...	...	...	6.889	19.900	20.645

In all of this study, the small difference between mean solar time and apparent solar time was ignored. Since New York is east of the 75th meridian, the mean sun is 1°2′ ahead of the position indicated by Eastern Standard Time. Although the position of the *mean* sun may be corrected to the position of the *true* sun through the equation of time, this correction was ignored.

Table B gives the ratio of direct solar radiation on a vertical surface to direct solar radiation on a horizontal surface for a north latitude of 40°46′, a west longitude of 73°58′ and a solar declination of 21°36′ (New York in mid-July).

The assumption with the least data to authorize it and, fortunately, the one with the least effect upon the result, is that the intensity of diffuse sky radiation on a vertical surface, regardless of orientation and sun position, is one-half of the intensity of diffuse sky radiation on a horizontal surface at the same time. The steps in finding the diffuse radiation incident upon a vertical surface corresponding to the observed total radiation on a horizontal surface equalled or exceeded on no more than 5 per cent of the days during the month of July (Table 2) are shown in Table C. The solar altitude at each hour is calculated on the assumption of a solar declination of 21°36′ (mid-July).

TABLE C—DETERMINATION OF DIFFUSE RADIATION ON VERTICAL SURFACE FROM TOTAL SKY AND SKY RADIATION ON A HORIZONTAL SURFACE

TIME (E.S.T.)	TOTAL SOLAR AND SKY RADIATION ON A HORIZONTAL SURFACE, BTU/HR FT² (FROM TABLE 2)	SOLAR ALTITUDE	RATIO OF DIRECT TO DIFFUSE RADIATION ON HORIZONTAL SURFACE (FROM FIG. 2)	DIFFUSE RADIATION INCIDENT UPON HORIZONTAL SURFACE BTU/HR FT²	DIFFUSE RADIATION INCIDENT UPON VERTICAL SURFACE (ASSUMED)
6 AM......	28	14° 40′	1.80	10.0	5.0
7	80	25° 47′	2.75	21.4	10.7
8	144	37° 7′	3.60	31.4	15.7
9	196	48° 21′	4.40	36.3	18.2
10	236	58° 56′	5.13	38.5	19.3
11	260	67° 29′	5.50	40.0	20.0
12	278	70° 49′	5.61	42.0	21.0
1 PM......	278	66° 31′	5.48	43.0	21.5
2	267	57° 33′	5.03	44.3	22.2
3	225	46° 50′	4.28	42.7	21.4
4	189	35° 33′	3.50	42.0	21.0
5	137	24° 14′	2.65	37.5	18.8
6	82	13° 10′	1.67	30.7	15.4
7	31	2° 38′	0.35	23.0	11.5

Some data for the observed intensities of diffuse sky radiation on vertical surfaces facing south, east and west were plotted by Houghten, Shore, Olson and Gunst in the A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 95. From these data, a suggested design curve for diffuse sky radiation on vertical surfaces was obtained (see Fig. C-1). This curve is compared with the data obtained in Table C. The diffuse sky radiation on a vertical surface depends upon the orientation of that surface with respect to the sun as well as the solar altitude and is not a constant fraction of the diffuse radiation on a horizontal surface at the same time. However, it is believed that the diffuse radiation assumed in preparing the sol-air temperature curves for vertical surfaces in this report is within about 10 Btu/hr ft² of the actual and the equivalent temperatures for vertical surfaces are not over 3 F in error, even for the case where the solar absorptivity is 1.0.

It should be noted that observed values of the total radiation incident upon a horizontal surface must always be broken down into direct and diffuse radiation before any attempt is made to *estimate* the total radiation incident upon a vertical surface and the trigonometric calculation will apply only to the *direct* radiation. For exam-

ple, assume the total solar radiation received on a horizontal surface at 6 p.m. in New York in mid-July is 82 Btu/hr ft² and that it is required to find the total radiation incident at that time upon a vertical wall facing west. The ratio of *direct* solar radiation on this vertical wall to the *direct* solar radiation on a horizontal surface at this time is 4.083. If one were to assume that this was also the ratio of *total* radiation, one might conclude that the total radiation incident upon the west wall would be 4.083 (82) or 323 Btu/hr ft². This is incorrect, however, because about 31 of the 82 Btu/hr ft² incident upon the horizontal is diffuse sky radiation; hence, the total radiation incident upon the west wall at this time is more nearly [4.083 (51) + 0.5 (31)] or 224 Btu/hr ft².

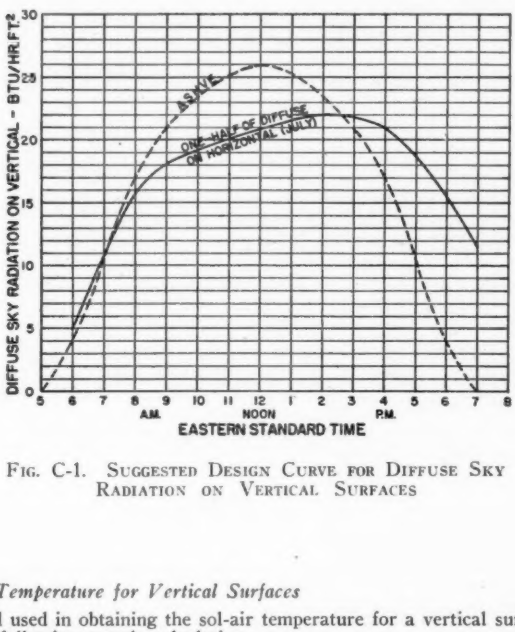

FIG. C-1. SUGGESTED DESIGN CURVE FOR DIFFUSE SKY RADIATION ON VERTICAL SURFACES

D. Sol-Air Temperature for Vertical Surfaces

The method used in obtaining the sol-air temperature for a vertical surface is illustrated by the following sample calculation:

At 9 a.m. (E.S.T.), the observed temperature of the outdoor air in the shade at the New York Meteorological Observatory is $t_a = 78$ F and the observed total solar and sky radiation incident upon the horizontal surface is $I = 200$ Btu/hr ft². The declination of the sun at this time is $D = 21°36'$. It is desired to find the sol-air temperature at this time for a vertical surface with an absorptivity of $b = 1.0$, facing *east.*

From Equation 4,

$$K_H = (\sin 21° 36')(\sin 40° 46') + (\cos 21° 36')(\cos 40° 46')(\cos 43° 58')$$
$$= 0.7472$$

Since $\sin H = 0.7472$, the solar altitude at this time is $H = 48°21'$.

From Fig. 2, the ratio of direct solar radiation to diffuse sky radiation for this solar altitude is 4.4. The diffuse sky radiation on the horizontal surface is 200/5.4 or

37 Btu/hr ft^2, while the direct solar radiation on the horizontal surface is 163 Btu/hr ft^2.

The azimuth of the sun at this time, A_s, may be found from Equation 8:

$$\sin (A_s - 180) = \frac{(\cos 21° \, 36')(\sin 43° \, 58')}{\cos 48° \, 21'}$$
$$= 0.9712, \text{ or}$$
$$A_s = 283° \, 47'$$

For a vertical surface facing *east*, the azimuth of the trace of the surface is $A_v = 360°$, and from Equation 7,

$$B = 360° - 90° - 283° \, 47' = -13° \, 47'$$

From Equation 6,

$$K_v = (\cos 48° \, 21')(\cos - 13° \, 47') = 0.6455$$

The intensity of *direct* solar radiation on the vertical surface facing east is:

$$I_v = \frac{0.6455}{0.7472} (163) = 140.8 \text{ Btu/hr ft}^2$$

The intensity of diffuse sky radiation on the vertical surface is assumed to be one-half of that on the horizontal surface or 18.5 Btu/hr ft^2.

The *total* solar and sky radiation incident on the vertical surface facing east is:

$$I = 140.8 + 18.5 = 159.3 \text{ Btu/hr ft}^2$$

For a solar absorptivity of $b = 1.0$, the sol-air temperature at this time for the vertical surface facing east is, from Equation 1,

$$t_e = 78 + \frac{159.3}{4} = 118 \text{ F}$$

E. Sol-Air Temperature for Glass

Assume that the fraction of incident total radiation which is transmitted directly through glass, the transmissivity, is designated by t; the fraction absorbed, absorptivity, by b; the fraction reflected, reflectivity, by r.

then,

$$t + b + r = 1$$

For the steady flow of heat through glass (no heat storage), it may be shown that the contribution to the cooling load per square foot of glass due to heat directly transmitted and heat conducted through the glass is:

$$\frac{q}{A} = \left(\frac{q}{A}\right)_t + \left(\frac{q}{A}\right)_c$$
$$= tI + U\left[(t_a - t_i) + \frac{bI}{h}\right] \quad \text{(9)}$$

where

$\frac{q}{A}$ = the contribution to the cooling load, Btu/hr ft^2,

I = the total solar and sky radiation incident upon the outdoor surface of the glass, Btu/hr ft^2,

U = the overall coefficient of heat transfer of the glass, Btu/hr ft^2 F,

h = the film coefficient of heat transfer at the outside surface of the glass, Btu/hr ft^2 F,

t_a = the temperature of the outdoor air, F,

t_i = the temperature of the indoor air, Fahrenheit degrees.

Note that

$$\frac{q}{A} = U\left[\left(t_a + \frac{bI}{h} + \frac{tI}{U}\right) - t_i\right] \quad \text{(10a)}$$

$$= U\,(t_e - t_i) \quad \text{(10b)}$$

The sol-air temperature for glass is:

$$t_e = t_a + I\left(\frac{b}{h} + \frac{t}{U}\right) \quad \text{(11)}$$

Unfortunately, the absorptivity, b, and the transmissivity, t, are not constant but vary with the angle of incidence of the sun's rays; this variation would have to be known for an exact application of this principle.

As an illustration of the use of Equation 11, consider the following case:

Assume the intensity of the incident solar and sky radiation is 300 Btu/hr ft^2 on a surface at normal incidence to the sun's rays; assume $t_a = 90$ F, $h = 4$ Btu/hr ft^2 F and $U = 1.13$ Btu/hr ft^2 F. For normal incidence of the sun's rays on ordinary window glass, $b = 0.04$ and $t = 0.88$. Then, the sol-air temperature for the glass in this case with the sun's rays at normal incidence is:

$$t_e = 90 + 300\left(\frac{0.04}{4} + \frac{0.88}{1.13}\right)$$

$$= 327 \text{ F}$$

The rate of heat transfer to the inside of the enclosure with $t_i = 80$ F, is, for this case,

$$\frac{q}{A} = 1.13\,(327 - 80) = 279 \text{ Btu/hr ft}^2$$

For engineering purposes, since the second term on the right side of Equation 9 is small in comparison with the first, the contribution to the cooling load due to heat transfer through glass is, approximately,

$$\frac{q}{A} = tI \quad \text{(12)}$$

In the foregoing example, the use of this approximation would give a result of 264 Btu/hr ft^2 instead of the correct result of 279 Btu/hr ft^2.

(See p. 106 for Discussion.)

1266

No. 1266

PREVENTING THE SOLUTION OF CO_2 IN CONDENSATES BY VENTING OF THE VAPOR SPACE OF STEAM HEATING EQUIPMENT

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This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with Carnegie Institute of Technology.

THIS PAPER presents the results of cooperative research studies [1] aimed at revealing, within practical limits, the possibilities of preventing the solution of CO_2 in condensates by venting of the vapor space of steam heating equipment. This work was undertaken in an attempt to discover another means of minimizing corrosion of condensate return lines when, perforce, CO_2-bearing steams must be used. It was predicated upon the previously published findings [2] of one of the authors which show that, other conditions being fixed, corrosion of condensate lines decreases as the dissolved CO_2 content of the condensate decreases.

General Findings

The general findings are that venting of the vapor space of steam condensers does not provide a means of producing CO_2-free condensates even when steams containing small concentrations of the gas are used. Notwithstanding, it is possible to produce thereby, from steams *rich* in CO_2, condensates containing gas concentrations of the same order of magnitude (2 to 4 ppm) as ordinarily are formed in equipment using steams of the *lowest* CO_2 content (*i.e.*, about 5 ppm) that are now generated from carbonate-bearing feedwater upon a commercial scale. Theoretically, therefore, venting provides a means of minimizing but not entirely preventing corrosion. Its general use, however, must await the commercial availability of suitable venting devices.

General Considerations

In 1939, Collins [2] published the curves reproduced in Fig. 1. These show that, other conditions being fixed, corrosion is proportional to the CO_2 concentration of condensates. The work of Mills and Urey [3] and their predecessors show that, at heater temperatures, equilibrium of CO_2 between steam and con-

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[1] Cooperative Research at Carnegie Institute of Technology, September, 1942, to December, 1943, sponsored by the A.S.H.V.E. Technical Advisory Committee on Corrosion.
[2] Corrosion in Steam Heating Systems, by L. F. Collins and E. L. Henderson. (*Heating, Piping & Air Conditioning*, Sept. 1939 to May, 1940, incl.)
[3] *Journal American Chemical Society*, Vol. 62, p. 1019, 1940.

Presented at the 51st Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, Mass., January, 1945.

densate can be attained in an extremely short period of time. At equilibrium, the CO_2 content of any condensate, free of alkaline entrainment, will be directly proportional to the partial pressure of CO_2 in the vapor phase contacting the condensate. In his latest publication, Collins [4] shows that in conventional designs of steam heating equipment, using CO_2-bearing steam, stratification of CO_2 and steam occurs and that the concentrations of CO_2 which

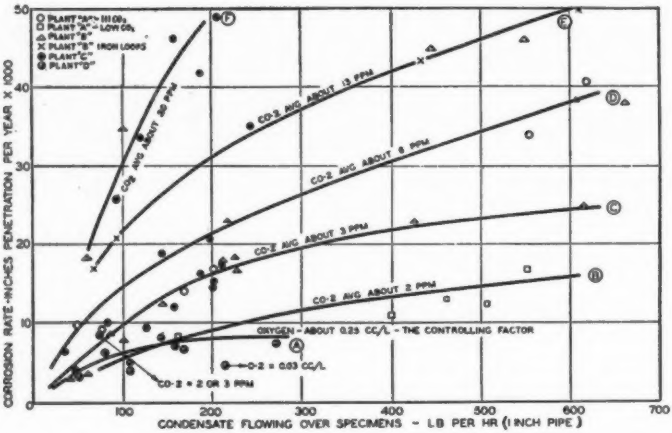

FIG. 1. CURVES SHOWING THE RELATION BETWEEN CORROSION AND GAS CONTENT OF CONDENSATE WHEN VARYING QUANTITIES OF CONDENSATE ARE FLOWING

eventuate may cause the hot condensate to dissolve an amount of CO_2 equal to that in the incoming steam.

From these observations it can be reasoned that, if CO_2 can be prevented from accumulating in the vapor space, lower partial pressures of CO_2 will result, less will dissolve in the condensate and, therefore, the latter will have a proportionately lower potential corrosivity. It was the aim of the present studies to determine the practical possibilities of venting gaseous CO_2 from the vapor space of heating equipment operating at high condensing rates, such as a water heater.

EXPERIMENTAL APPROACH

In the design of all heating equipment, the principal objective sought is to achieve the ultimate in thermal economy. Normally, steam and condensate are caused to flow in the same direction. In addition, in some units, wherein water

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[4] Studies of the Mechanism of Solution of CO_2 in Condensates Formed in the Steam Heating Systems of Buildings, by Leo F. Collins. (See p. 39, this volume.)

is the cooling medium, steam and condensate are caused to flow counter current to the cooling water as a result of which *undercooling* of the condensate occurs.

In view of the principles set forth in the preceding paragraph, it will be recognized that the design features cited favor solution of CO_2 in the condensate. If this is true, then it should follow that, if steam and condensate are made to flow in opposite directions and if the CO_2, which accumulates at the end of the steam path, is vented to the atmosphere, conditions favoring solution of a minimum of CO_2 in the condensate should be produced. The present studies were conducted in such a manner as to test these postulates.

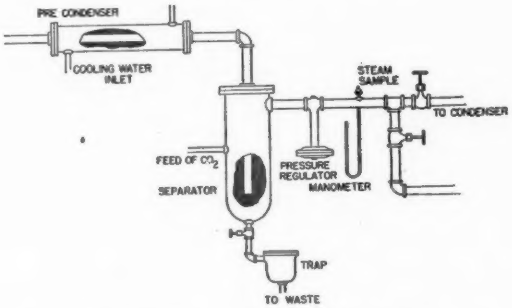

FIG. 2. ARRANGEMENT OF EQUIPMENT PRECEDING EXPERIMENTAL CONDENSERS

TEST TECHNIQUE

Steam from a main supplying the laboratory building was used throughout the tests. To minimize the interference of entrained boiler water salines, and to facilitate enrichment of the steam with CO_2 when desired, the equipment preceding the condensers was arranged as shown in Fig. 2.

In all tests, to prohibit the escape of undissolved gas along with the condensate, the condensate line was kept flooded at a predetermined level. This was accomplished automatically by means of a *home-made* constant-level float valve. Essentially, this device consisted of a glass float linked to a carefully machined brass cylinder. As the latter raised or lowered, it throttled the exit port in the condensate line.

Condenser No. 1

This unit was a water heater of the helical coil-type, made entirely of copper alloy. The exterior surface of the coil and the interior surface of the jacket were heavily tinned to eliminate the possible effects of corrosion products. It had a rated capacity of 180 gal per hour when utilizing steam at atmospheric

pressure, with an inlet water temperature of 40 F and an outlet water temperature of 140 F, *i.e.*, it contained about 3.5 sq ft of heating surface.

The first series of tests were made with steam and condensate flowing in the same direction and counter-current to the cooling water, *i.e.*, conventional operation. A steam pressure of 3 psi gage was maintained, and the cooling water was regulated so as to cause condensation of about 225 lb of steam per hour.

The vent and subsequent venting apparatus were arranged as shown in Fig. 3. With these arrangements, data were collected while venting steam at

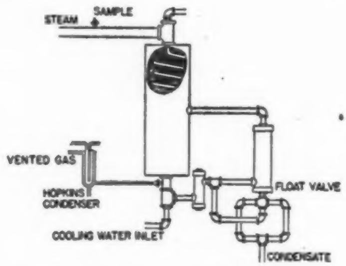

FIG. 3. ARRANGEMENT OF EXPERIMENTAL CONDENSER No. 1

different rates up to 1.81 per cent of that entering the heater, with the incoming steam containing about 4 ppm of CO_2 and also after it had been enriched to contain about 40 ppm of the gas.

The data collected are summarized in Table 1. The values shown are an average of 4 readings taken at consecutive 15-min intervals after the unit had operated sufficiently long, at a given venting rate, to reach a steady state.

TABLE 1—AVERAGE RESULTS OBTAINED WITH HEATER No. 1

Condensing Rate Lb per Hr	CO_2 ppm in			Venting Rate[a]	Per Cent of Incoming CO_2 Measured in	
	Incoming Steam	Vent Steam	Condensate		Condensate	Vented Steam
238	3.6	134	1.7	1.73	42.6	57.4
222	3.0	163	1.4	1.25	45.7	54.3
241	3.8	252	2.3	0.90	49.0	51.0
226	3.3	407	1.8	0.43	47.2	52.9
232	39	1886	6.2	1.81	15.7	84.3
232	44	3500	8.2	0.98	19.3	80.7
225	37	3300	6.1	0.94	16.6	83.4
238	33	5290	8.0	0.52	23.9	76.1
236	44	6940	11.1	0.43	27.1	72.9

[a] As per cent of incoming steam.

When an attempt was made to operate this unit with counter-flow of steam and condensate, it was found impossible to condense as much steam as in the first series of tests. Because of this experimental difficulty it was decided to cease work with this unit and to begin the studies on Condenser No. 2.

Condenser No. 2

This unit was a water heater of the hair pin tube type. The tubes were of copper alloy, and the shell was of steel. Prior to starting the test, all the inside

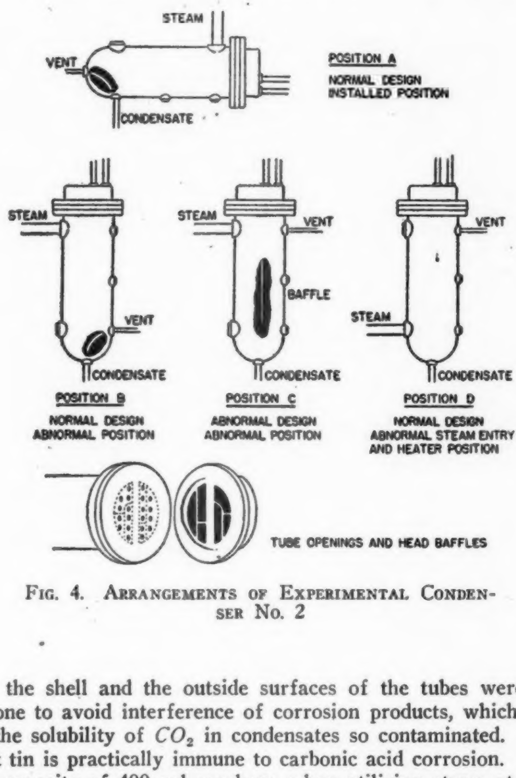

FIG. 4. ARRANGEMENTS OF EXPERIMENTAL CONDENSER No. 2

surfaces of the shell and the outside surfaces of the tubes were tin plated. This was done to avoid interference of corrosion products, which conceivably may affect the solubility of CO_2 in condensates so contaminated. It has been shown [5] that tin is practically immune to carbonic acid corrosion. This heater had a rated capacity of 400 gal per hour when utilizing steam at atmospheric pressure, with an inlet and outlet water temperature of 40 and 180 F respectively, *i.e.*, it contained 9 sq ft of heating surface.

In all the tests involving Unit No. 2, a vent-condenser [6] was used which differed in one respect from the design used with Unit No. 1. Why this

[5] More Information on Corrosion in Steam Heating Systems, by L. F. Collins. (Proceedings 4th American Water Conference *Engineers' Society of Western Pennsylvania*, November, 1943.)

[6] In all tests, a modified Hopkins Condenser was used as a vent condenser. This is a small glass condenser common to chemical laboratories.

change was made is described later in the paragraph entitled *Criteria for the Design of Venting Devices.*

With the heater positioned and/or arranged in the four different ways, depicted by the sketches in Fig. 4, tests were made at condensation rates corresponding to 300 and 600 lb of steam, approximately, per hour at 3 psi gage. For each condensation rate, tests were made at CO_2 levels in the incoming steam of 30 and 60 ppm respectively. For each CO_2 level, tests were run at rates of venting corresponding to 0.25, 0.50, and 1.0 per cent of the weight of steam entering the heater in unit time.

In every test, the following general procedure was used:

The heater was put into operation and adjustments were made to give the desired condensation rate. The CO_2 feed to the incoming steam was set at a predetermined value, as was also the venting rate. When the system had reached a steady state (usually in about one-half hour) as shown by preliminary checks, readings were taken. The data hereinafter presented represent the average of the values obtained from three or more consecutive readings made at about 20-min intervals. Thereafter, the venting rate was changed to a lower predetermined value and the same procedure repeated.

Normal Design and Installed Position—(A): With the unit installed in a horizontal position (arrangement A, Fig. 4), *which is its normal position in actual practice*, it seems permissible to assume that, as the condensate from the upper elevation gravitates downward, it is likely to contact cooler tubes. This, coupled with the fact that at the lowest level the gas phase richest in CO_2 persists, should make for maximum solution of the gas in the condensate. The heater in question was provided with a vent, as shown by arrangement A of Fig. 4 and tested under these conditions. The average results obtained are given in Table 2.

Normal Design—Abnormal Position—(B): In this series the heater's original design was not modified, but the unit was positioned so the tubes were vertical, *i.e.*, arrangement B of Fig. 4. With such an arrangement, it seems improbable that undercooling of the condensate can take place. Under these conditions, the average results obtained are those given in Table 3.

TABLE 2—AVERAGE RESULTS—HEATER NO. 2—NORMAL DESIGN AND INSTALLED POSITION

Condensing Rate Lb per Hr	CO_2 ppm in			Venting Rate[a]	Per Cent of Incoming CO_2 Measured in	
	Incoming Steam	Vent Steam	Condensate		Condensate	Vented Steam
630	61	4035	13	1.08	24.4	76.9
630	56	7550	17	0.50	30.4	69.0
606	55	11158	23	0.27	42.0	56.0
614	34	2108	11	1.11	32.4	69.1
614	33	3357	14	0.54	43.0	56.7
612	33	5894	15	0.25	45.4	54.6
366	59	4728	15	0.95	25.4	74.9
366	58	12148	17	0.35	28.6	72.3
366	58	18420	17	0.24	29.6	70.8
370	29	2470	11	0.75	37.6	62.1
366	31	4605	10	0.45	31.0	67.8
366	29	9142	9	0.22	31.0	66.9

[a] As per cent of steam entering unit.

TABLE 3—AVERAGE RESULTS—UNIT NO. 2—NORMAL DESIGN—ABNORMAL POSITION

CONDENSING RATE LB PER HR	CO_2 PPM IN			VENTING RATE[a]	PER CENT OF INCOMING CO_2 MEASURED IN	
	INCOMING STEAM	VENT STEAM	CON-DENSATE		CON-DENSATE	VENTED STEAM
575	56	3090	7	1.00	13.1	87.6
552	56	7324	9	0.70	15.8	84.5
552	60	10416	15	0.43	25.0	75.6
552	54	16078	22	0.17	40.6	55.9
578	30	2548	5	1.04	16.0	84.4
576	32	4664	9	0.61	24.2	75.5
576	39	7008	10	0.38	25.6	73.4
576	32	10748	13	0.19	41.2	59.0
315	55	4621	7	1.11	12.4	86.1
321	56	10757	12	0.42	20.6	80.3
315	54	13685	17	0.27	31.8	68.3
330	33	3130	7	0.93	20.9	79.9
330	31	4826	11	0.46	34.5	65.6
330	32	7485	14	0.24	43.6	56.6

[a] As per cent of steam entering unit.

Abnormal Design—Abnormal Position—(C): For this series, the heater's design was modified to the extent that a baffle was installed to direct the path of the steam, as shown in arrangement C of Fig. 4, and the vent was relocated, as is also shown in the sketch. By this arrangement, undercooling of the condensate was prevented, and the CO_2 which normally accumulates at the bottom was forced to the top of the unit. Under these conditions, the average results obtained are those given in Table 4.

TABLE 4—AVERAGE RESULTS—UNIT NO. 2—ABNORMAL DESIGN—ABNORMAL POSITION

CONDENSING RATE LB PER HR	CO_2 PPM IN			VENTING RATE[a]	PER CENT OF INCOMING CO_2 MEASURED IN	
	INCOMING STEAM	VENT STEAM	CON-DENSATE		CON-DENSATE	VENTED STEAM
550	59	6150	2	1.15	3.5	97.1
555	59	10630	3	0.55	5.1	95.0
560	57	21172	4	0.29	6.9	93.0
606	36	2900	4	1.07	11.1	89.0
579	35	5995	5	0.54	14.2	86.0
598	35	11546	6	0.26	17.2	82.4
300	60	5741	4	1.05	6.7	93.1
300	59	11233	5	0.53	8.5	92.7
300	60	21710	6	0.24	10.0	89.3
316	30	2778	4	1.00	13.3	87.1
310	30	5422	4	0.51	13.3	87.2
310	31	9648	6	0.25	19.3	81.0

[a] As per cent of steam entering unit.

Normal Design—Abnormal Steam Entry and Heater Position—(D): In this series the steam was caused to enter the unit close to the normal condensate level, and the vent was located at the top of the unit, as shown in arrangement D of Fig. 4. This, in effect, was the same as the arrangement in the tests of series C but accomplished without installing the baffle. With this arrangement, the average results obtained are those given in Table 5.

VENTING EFFICIENCIES

A comparative study of the data from the five series of tests indicates clearly that control of the CO_2 content of condensates, formed from and in contact with CO_2-bearing steams, by venting of the vapor space is limited and conditioned by:

1. The amount of steam vented in unit time.
2. The CO_2 content of the incoming steam.
3. The proper location of the bleed point.
4. Design characteristics of the unit in question.

Optimum Percentage of Venting

There are well informed engineers who contend that the venting of steam heating equipment, as a means of thwarting corrosion, is predestined to be impractical. Their capital argument is that the problem of disposing of the vented steam will be as costly and involved as the replacement of those parts which normally fail due to corrosion.

It will be shown later that, conceivably, there are rather simple ways for disposing of small amounts of vented steam. It will also be shown that, when intelligently employed, venting of small amounts of steam effectively minimizes the CO_2 content of condensates. Accordingly, it is believed that the

TABLE 5—AVERAGE RESULTS—UNIT NO. 2—ABNORMAL STEAM ENTRY AND HEATER POSITION

Condensing Rate Lb per Hr	CO_2 ppm in			Venting Rate[a]	Per Cent of Incoming CO_2 Measured in	
	Incoming Steam	Vent Steam	Condensate		Condensate	Vented Steam
624	58	4863	4	1.01	6.9	92.5
614	57	9193	4	0.55	7.0	93.1
608	55	17328	6	0.27	10.9	89.1
612	31	2517	3	1.05	9.7	90.2
614	30	4700	3	0.53	10.0	89.8
612	27	8923	4	0.26	14.8	85.4
310	60	6136	3	1.02	5.0	95.3
310	61	11778	4	0.51	6.6	94.0
310	60	21788	5	0.27	8.3	92.0
328	28	2714	2	0.96	7.1	92.4
324	28	5227	3	0.48	10.7	89.3
324	30	10789	6	0.25	20.0	79.6

[a] As per cent of steam entering unit.

present studies outline a method for control of carbonic acid corrosion that heretofore has not been available for use by steam utilization engineers.

Prior to the inauguration of these studies it was decided, quite arbitrarily, that for venting to be practical, not more than about two per cent of the steam entering a given condenser could be so dissipated. Thus it is that all tests were made at venting rates below two per cent.

In Figs. 5, 6 and 7, inclusive, are plotted the amounts of CO_2 found in the condensates (as per cent of that in the incoming steam) against venting rates,

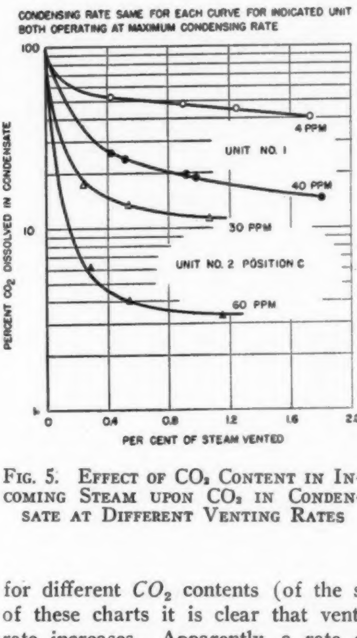

FIG. 5. EFFECT OF CO_2 CONTENT IN INCOMING STEAM UPON CO_2 IN CONDENSATE AT DIFFERENT VENTING RATES

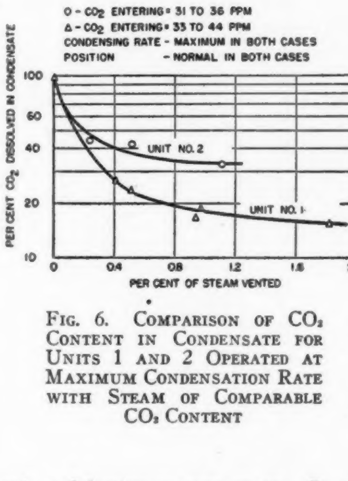

FIG. 6. COMPARISON OF CO_2 CONTENT IN CONDENSATE FOR UNITS 1 AND 2 OPERATED AT MAXIMUM CONDENSATION RATE WITH STEAM OF COMPARABLE CO_2 CONTENT

for different CO_2 contents (of the steam) and heater arrangements. In all of these charts it is clear that venting efficiency[7] decreases as the venting rate increases. Apparently, a rate equal to about one-half per cent of the steam entering a given unit is the most efficient, providing venting is done from the optimum bleed point.

In Figs. 5, 6 and 7, plotting the value of 100 per cent solution of the gas at zero venting rate is not based upon an academic postulate. Tests of both heaters showed that this occurs. When unit No. 1 had reached a steady state, the vent was suddenly closed. Under these conditions, complete solution of the gas was brought about in about 30 min. When unit No. 2 (arranged as shown by A of Fig. 4) had reached a steady state, at a venting rate of 1.35 per cent, closing of the vent brought about complete solution of the gas in about 20 min at a condensing rate of 600 lb of steam per hour.

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[7] Since the per cent of CO_2 removed is not a straight line function of the per cent of steam vented, doubling the venting rate does not halve the per cent CO_2 remaining in the condensate.

Effect of CO_2 Concentrations in Incoming Steam

Venting efficiency decreases as the CO_2 content of the incoming steam decreases. For CO_2 content of steam below about 4 ppm, venting is appar-

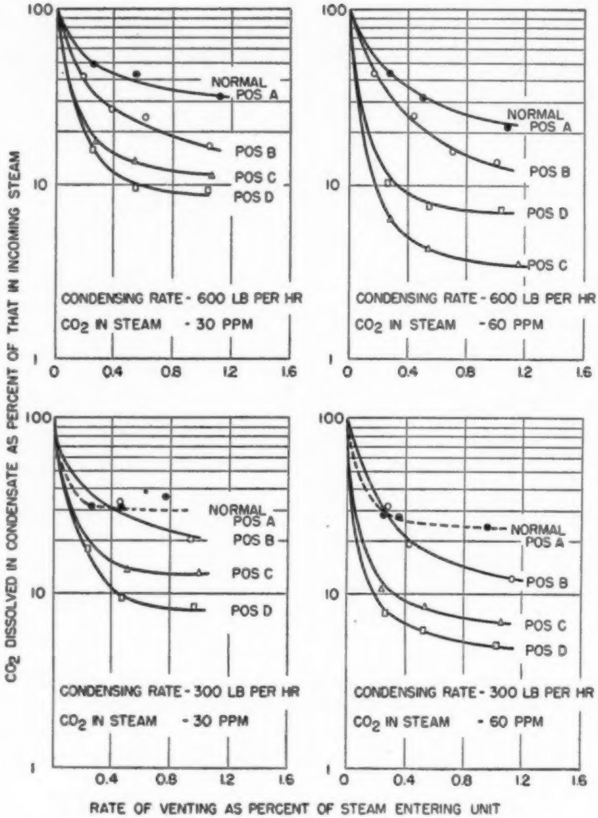

FIG. 7. EFFECT OF FOUR DIFFERENT ARRANGEMENTS ON UNIT No. 2 WHEN OPERATED AT TWO CONDENSATION RATES AND TWO CO_2 LEVELS

ently of little practical value. The latter observation is attested by the curves in Fig. 5. It should be noted that the data for unit No. 2, plotted therein, which pertain to series C, are practically identical with the results obtained with series D.

Optimum Bleed Point

Obviously, the optimum bleed point is at that location where the highest CO_2 concentration exists. While ordinarily this will be close to the condensate level, *i.e.*, in the bottom of the unit, it may be elsewhere, depending upon the direction and path of steam flow. The point of highest CO_2 concentration is always at the end of the steam path. It was for these reasons that in the tests of series A and B venting was from the bottom, whereas in series C and D it was from the top of the experimental unit.

Initially, there is a temptation to postulate that CO_2 accumulates at the bottom of conventional types of steam-condensing apparatus because of the differences in density between steam and CO_2. That such accumulations are due to steam flow rather than to differences in gas densities is attested by the findings of series A and B as contrasted with those of series C and D. The data in Tables 4 and 5 show higher CO_2 values for the vented steam than in Tables 2 and 3, all other conditions being equal. It is inconceivable that this could have occurred if gas densities were the controlling factor.

With the unit arranged as shown by B of Fig. 4, which is essentially an orthodox arrangement, the vent was suddenly closed so that the gas could accumulate. When equilibrium became established, *i.e.*, the amount of CO_2 dissolved in the outgoing condensate equalled that entering with the steam, sufficient heating surface had been blanketed with gas to reduce the steam condensing rate slightly over one per cent. With the arrangement shown by D of Fig. 4, which is an unorthodox arrangement, when equilibrium had become established, the steam condensing capacity had been decreased about 30 per cent. Certainly, therefore, an arrangement such as is shown in D is definitely impractical unless a suitable venting apparatus is provided.

Effects of Design Characteristics

The curves in Fig. 6 contrast the CO_2 dissolved in the condensate for the two units studied, when both were operated in an orthodox manner at maximum condensation rates and with steams of comparable CO_2 contents. The better performance of unit No. 1, it will be shown presently, resulted from the fact that undercooling of the condensate did not take place as is the case for the values relating to unit No. 2.

The curves in Fig. 7 show the average results obtained with unit No. 2 when it was arranged in the four different ways, depicted by the sketches in Fig. 4 but operated at two different condensation rates and two CO_2 levels for the incoming steam.

These charts demonstrate clearly (1) the effects of undercooling of the condensate, and (2) the value of utilizing steam scavenging.

In a unit of this design, arranged in its normal position, the condensate formed at the upper elevations is likely to contact progressively cooler tubes. This favors *undercooling* of the condensate and should become more pronounced as the rate of condensation increases. It has been shown, too, that with such an arrangement, the vapor in contact with the undercooled condensate is richest in CO_2; that both of these factors make for maximum solution of CO_2 under otherwise identical conditions. That this occurred is clearly shown by the data plotted in Fig. 7. That undercooling takes place more at high condensation rates is similarly shown.

The effects of undercooling can be demonstrated, too, in another manner. By comparing arrangements A and B it can be visualized that in arrangement B the condensate drained quickly from the vertical tubes, while in arrangement A it flowed from one tube to another. Thus it should follow that other conditions being equal, a given CO_2 content in the gas phase should bring about less solution of CO_2 in the condensate with arrangement B than with arrangement A. The curves in Fig. 8 show the relationship found to exist. It is interesting to note that the values for unit No. 1 subscribe to the lower curve. Apparently in this unit undercooling did not take place.

In comparing the results obtained by the two methods of steam scavenging it is seen that, except in one series, bringing the *purest* steam into contact

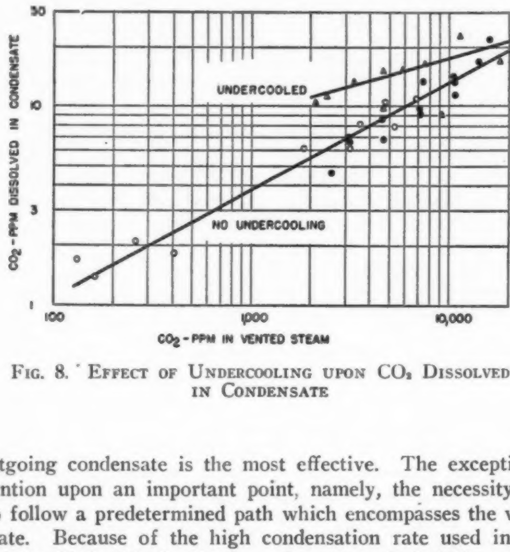

FIG. 8. EFFECT OF UNDERCOOLING UPON CO_2 DISSOLVED IN CONDENSATE

with the outgoing condensate is the most effective. The exception, however, focuses attention upon an important point, namely, the necessity for causing the steam to follow a predetermined path which encompasses the whole area of the condensate. Because of the high condensation rate used in the tests of series D and the high CO_2 in the incoming steam it is believed a *secondary pocket* of CO_2 accumulated, thus curtailing the effectiveness of the scavenging steam. This same phenomenon has been experienced in early design of deaerating types of industrial boiler feedwater heaters.

CRITERIA FOR THE DESIGN OF VENTING DEVICES

Practically, venting is impossible with equipment operating at subatmospheric pressures. On the other hand, a system in which the condensing equipment operates at positive pressures and the return lines under negative pressures would appear to yield most readily to the employment of venting.

In building steam heating equipment it rarely happens that a single piece of equipment is found which condenses more than about 1000 lb of steam per hour. If venting of about one-half per cent of the incoming steam represents

the optimum venting rate, then disposal of not more than five pounds of condensate per hour comprises the mechanics of the problem of venting.

To be practical, the venting device should separate the noncondensable gases from the condensate formed in the venting device and purge them to the atmosphere; returning the condensate, containing a negligible amount of dissolved gas, to the main return line.

Throughout the early experiments with unit No. 1, the modified Hopkins condenser was constructed as shown by arrangement A of Fig. 9. With this design the vented condensate immediately contacted the *cold finger* and tended to *hold* CO_2. For this reason the condenser was further modified as shown in arrangement B of Fig. 9. In this design the condensate in the venting

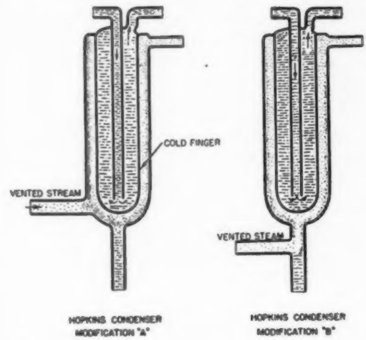

FIG. 9. CONSTRUCTION OF MODIFIED HOPKINS CONDENSER USED IN EARLY EXPERIMENTS WITH UNIT No. 1

apparatus is continuously *boiled* by the incoming steam. The gas content of the condensate leaving both types of apparatus, under otherwise identical conditions, is typified by the data in Table 6. Clearly, the modified design B was a distinct improvement.

It is believed that condensate of the quality typified by the data for design B may safely be returned to the main return line. While its CO_2 content will be comparatively high, the amount of condensate produced is so small that the

TABLE 6—DATA COMPARING THE PERFORMANCE OF TWO MODIFIED HOPKINS CONDENSERS

Steam Condensed Lb per Hr	Modification A CO_2 ppm in Steam to Condenser	Condensate Leaving Condenser	Steam Condensed Lb per Hr	Modification B CO_2 ppm in Steam to Condenser	Condensate Leaving Condenser
4.2	1886	12	..	...	...
2.8	3500	35	3.1	2714	28
1.2	5290	375	1.6	5227	100
1.0	6940	967	0.8	10789	156

increase in CO_2 concentration of the aggregate in the main return line will not be of practical importance.

In the design of a venting device which utilizes water as the cooling medium, provision should be made to guarantee no undercooling of the condensate. A device made of finned-type tubing, thus using air as the cooling medium, would appear preferable for most services. In Fig. 10 a possible design is suggested.

(1) Steam inlet
(2) Steam outlet orifices
(3) Tubing ⅛ in. diameter
(4) Condensate outlet-orificed-to main return or waste
(5) Threaded connection
(6) Splash plate
(7) Fluted tubing
(8) Sludge pocket
(9) Shield
(10) Outlet for gases

Materials of Construction: 18–8 stainless steel, aluminum, or tin plated copper alloys. (Same material should be used throughout.)

Note: Unit must be mounted in vertical position. Not usable on condensers operated at negative pressures.

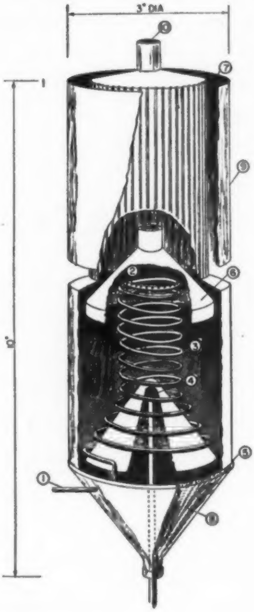

FIG. 10. ONE POSSIBLE DESIGN OF AN AIR-COOLED VENT CONDENSER

Insofar as the authors can determine few, if any, of the so-called air vent valves operated thermostatically, appear suitable for venting CO_2. The apparent reason is that the accumulation of CO_2 necessary to induce solution of important quantities of CO_2 does not depress the temperature of the steam-gas mixture sufficiently to cause the valves to open.

SUMMARY

It has been shown that by intelligent venting of the vapor space of steam condensing equipment, steams rich in CO_2 can be caused to produce condensates containing small amounts of CO_2, i.e., of the order of 2 to 4 ppm; and that venting is of little practical value when the incoming steam contains less than

about 5 ppm of CO_2. It has been shown elsewhere [8] that where boilers must use important quantities of carbonate-bearing feedwater, the inadequacy of water processing methods now in use preclude the commercial production of steam containing less than such quantities of CO_2; and that CO_2 concentrations in the condensate of the order of 2 to 4 ppm are capable of causing corrosion trouble if other contributing factors are optimum.

In view of these facts, it is clear that venting provides a means of mitigating but not entirely preventing corrosion troubles due to carbonic acid. As such, it is an expedient which lies definitely within the province of the steam utilization engineer and its effectiveness is comparable to the water processing methods available to engineers in charge of steam production.

Managements should appreciate that, with the means now available, the efforts of both the steam utilization engineer and the steam production engineer are limited to minimizing rather than to preventing corrosion when perforce the steam generator uses important quantities of carbonate-bearing feedwaters.

Apparently the ultimate solution of the CO_2 corrosion problem must await the development of suitable chemicals which can be added to the steam to neutralize carbonic acid, or the development of methods for processing carbonate-bearing feedwaters so as to completely eliminate the entrainment of CO_2 with the steam. Studies dealing with neutralizing chemicals, especially those of an organic nature, are in progress elsewhere.[9] Studies dealing with the development of improved water processing methods are under consideration by the A.S.H.V.E. Technical Advisory Committee on Corrosion.

Acknowledgments

To the engineers and chemists, and there were many who had no intimate concern with these studies but who, nevertheless, volunteered many helpful suggestions and much sound advice, the authors are indebted. Acknowledgment is due to Dr. J. C. Warner, head of the department of chemistry, Carnegie Institute of Technology, for his patience and counsel particularly in the early days, and to Dr. J. P. Fugassi for valuable assistance and suggestions. The generosity of G. D. Winans, engineer of steam distribution, The Detroit Edison Co., in loaning the experimental units, and of C. H. Fellows, head of the chemistry division, Research Department of the same company, in loaning an evolution carbonate determination apparatus, is also acknowledged.

APPENDIX

Concerning Determination of Carbon Dioxide

In these studies, the determination of CO_2 involved the analysis of samples over an extremely wide range of concentrations. Because the accuracy of the methods commonly used to determine CO_2 have been for some time the source of considerable conjecture amongst analytical chemists, it seems entirely proper to record the experiences gained during these studies wherein several methods were used. Such is the purpose of this appendix.

[8] Loc. Cit. Note 2.
[9] RI Report by A. A. Berk (U. S. Bureau of Mines, 3754, June, 1944).

Undissolved Gas

In the first series of tests (Heater No. 1), the steam supply contained 3 to 4 ppm of CO_2 and considerable air.[10] As a result, the CO_2 in the noncondensable gases vented from the unit consisted of only approximately 20 per cent CO_2. When the noncondensable gases were collected in a eudiometer tube over sulphuric acid (0.5 per cent solution) and subsequently analyzed in a Hempel apparatus, satisfactory balances were obtained. When, however, the steam was enriched so as to contain 40 ppm of CO_2—thus causing the noncondensable gases to contain about 75 per cent CO_2—this method did not give good balances. It is believed the inaccuracies resulted from solution of CO_2 in the acid.

Probable Accuracy of the Analytical Data

In the tests involving steams containing CO_2 of the order of 4 ppm, no data were accepted in which the CO_2 leaving the unit via the vent plus that dissolved in the condensate differed from that entering the unit by more than 20 per cent. For CO_2 values in the incoming steam of a higher order, data differing by more than 10 per cent were rejected. Better precision could not be hoped for because of the inherent limitations of the methods available for the determinations of small concentrations of dissolved CO_2. Since the CO_2 entering the experimental condensers was determined by a method entirely different from that used in measuring the outgoing CO_2, the possibility of accidental balances due to consistent errors is precluded.

By passing the vented gases (leaving the modified Hopkins condensers) through a magnesium perchlorate filled drying tube and absorbing the CO_2 in Ascarite, satisfactory balances were obtained for CO_2 concentrations within the range 2000 to over 20,000 ppm.

Dissolved CO_2

The *ASTM* evolution method was found satisfactory but time consuming on condensates containing up to about 1200 ppm of CO_2. On condensates not otherwise contaminated equally good results were obtained when an excess of standard $Ba(OH)_2$ was added to a 200 cc sample and the excess subsequently back-titrated with standard hydrochloric acid using phenolphthalein as indicator.

When alkaline materials were present, such as results from the entrainment of boiler water salines, it was found that titration of samples between pH 8.5 to 5.0, using a double indicator solution of o-cresolphthalein and methyl red, gave satisfactory CO_2 balances.[11] In retrospect, however, it is believed preferable to determine the bulk of the CO_2 content of steam by separation of the gas, using the Hopkins condenser modification B, and subsequent absorption of it in Ascarite. The relatively small amount of CO_2 in the condensate from the condenser can be determined by titration.

DISCUSSION[12]

L. F. COLLINS: In searching for a capable discusser of these two papers we were fortunate in finding that a few years ago some comparable research work of a laboratory nature was done by the Consolidated Gas and Electric Light and Power Co., Baltimore. Dr. Guernsey, who was responsible for the work, was kind enough to give both Mr. McKinney and myself a copy of the results, but as the manuscript received indicated that it was confidential information we did not feel at liberty to cite work to which we had privileged access. Since that time, however, it has been

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[10] Because the Boiler feedwater, from which the steam was derived, is not deaerated.

[11] Determination of Carbon Dioxide in Water, by D. S. McKinney and A. M. Amorosi. (Industrial & Engineering Chemistry, Vol. 16, p. 315, May, 1944.)

[12] This discussion also covers Chapter No. 1265, p. 39.

decided that the results can be made public and therefore Dr. Guernsey will refer to them.

E. W. GUERNSEY, Baltimore, Md. (WRITTEN): The authors of these two papers deserve the thanks of all those concerned with corrosion in heating systems for the large amount of conscientious effort which has gone into the work described. They have made a substantial addition to the experimental data concerning the mechanism of entry of carbon dioxide into condensate and the means for keeping dissolved carbon dioxide at a minimum.

It may, or may not, be found generally feasible to vent a small part of the steam from heating equipment to reduce corrosion. In any case, the data presented have a very practical value in that they contribute to the fuller understanding of the factors which influence the solution of carbon dioxide and other gases in the condensate. Such an understanding should make it possible to take this factor into account in designing equipment. It will also often make possible the intelligent appraisal of the tendency toward corrosion to be expected with particular equipment or methods of operation.

The papers which have been presented afford additional assurance that there are no important unrecognized factors affecting the solution of gases in steam-heated equipment. We have, over a period of years, studied the tendency toward corrosion in various types of steam-heating equipment and have been able, in most instances, to rationalize the finding of high concentrations of carbon dioxide in condensate, and accelerated corrosion in particular systems, and in many cases to indicate the direction in which to seek improvement. This could be done by considering the method of operation of the equipment and the pattern of steam and condensate flow, and by applying the same principles employed in the two papers which have been presented. In addition, some laboratory studies were made under conditions permitting a better control of variables than was possible in the field. These results were described and given limited circulation in 1938, but were not published. That work and the studies which have been presented are in many respects supplementary. For that reason, it may be worth while to recall briefly some of our conclusions.

In the matter of venting, we sought first to establish the theoretical limitations of this practice. For a hypothetical case in which each element of steam is assumed to move forward, in contact with its own condensate, with the contained gases distributed in equilibrium between steam and condensate and with a final small portion of uncondensed steam withdrawn, it is of course possible to calculate from data on the solubility of carbon dioxide the fraction of contained carbon dioxide which would enter the condensate. The effectiveness of venting steam, calculated for such an ideal case, is shown in Fig. A.

Note that with steam condensing at atmospheric pressure the elimination of about 98 per cent of the carbon dioxide by venting one per cent of the steam is theoretically possible. Other calculations not shown were made for higher steam pressures. At 40 lb per square inch, gage, the theoretical elimination would be 95 per cent with one per cent purge.

Note further, however, that the effectiveness of venting is greatly reduced if the steam removed is not at the very end of the path of the steam flow in the apparatus. If as little as 0.1 per cent of the steam condenses *downstream* from the point of venting, the theoretical elimination with one per cent purge drops to 89 per cent, and with one per cent of such *downstream-condensation* the maximum elimination is only 50 per cent. This suggests that venting would lose its effectiveness unless obtained from a point just over the condensate level. This practice would be difficult except in connection with a type of trap which maintains a definite level of condensate.

It is, of course, not to be expected that carbon dioxide would, in practice, be eliminated by purging with the effectiveness calculated for the ideal case, at least not with equipment in which condensate and steam flow in the same direction. It was established experimentally, however, that the conclusions from the calculation were essen-

tially correct in that a high proportion of the carbon dioxide could be eliminated with a small fraction of steam *bled*. For this purpose a laboratory condenser such as is illustrated in Fig. B was used. This is simply a one-inch water-cooled tube B, terminating in a vertical ⅜-in. tube C, acting as a condensate receiver. Steam was supplied at A. Manual operation of valve D maintained the desired condensate level, as indicated on level indicator G, in the receiver C, and thus insured against any leaking of steam with the condensate. The vented steam was taken from the space

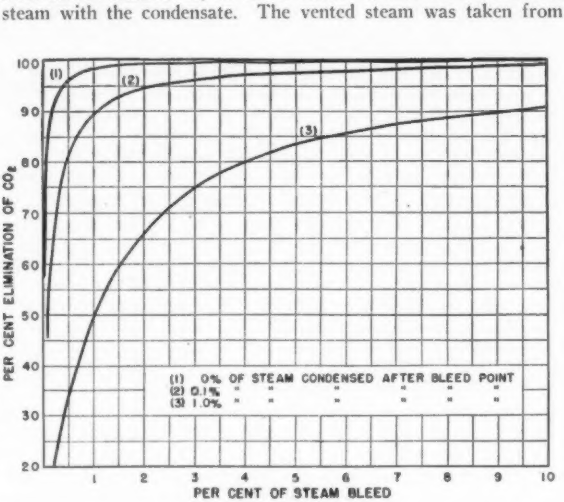

Fig. A. Elimination of Carbon Dioxide by Bleeding at Atmospheric Pressure

above the condensate through the line E. Condensate was withdrawn for analysis through the cooling coil H and valve F. Typical results are shown in Table A.

Note that the actual elimination varies from 91 to 93 per cent, as compared to the calculated values for the ideal case of 96 to 99 per cent. This, then, establishes that, as predicted by the theory, a high elimination was possible in apparatus of this particular design. It did not indicate necessarily what it might be possible to do with equipment of commercial design. The observations presented by Mr. McKinney show, however, that with a commercial water heater of conventional design, a substantial elimination can be effected by properly arranged purging means.

Table A—Elimination of Carbon Dioxide by Venting

Pressure, lb/in.²	Per Cent Bled	Per Cent Elimination of CO_2	
		Calculated	Observed
40	1.2	96.0	90.6
40	2.6	98.1	91.2
40	3.8	98.7	92.4
8	2.1	98.8	93.1

The applicability of venting for preventing corrosion in many other types of steam heating equipment is not necessarily indicated, and, in fact, it is to be expected that venting will not be practicable in some cases. As one difficulty, the necessity for locating the vent at the end of the path was mentioned and this has also been stressed in the papers presented. This point is qualitatively shown by the results of some experiments with the apparatus of Fig. B modified by the removal of the insulation from the vertical condensate receiver, C, thus allowing it to act as an air condenser.

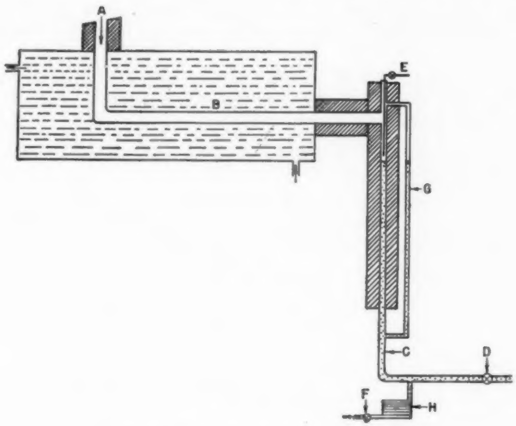

FIG. B. EXPERIMENTAL CONDENSER

denser. The condensate level was held at various distances below the vent point and the effect on the elimination of carbon dioxide noted. The results are shown in Table B.

The unknown but presumably small amount of *downstream-condensation* occurring in 19.5 in. of receiver tube reduced the elimination with 1.1 per cent purge to 49 per cent, as compared with 92 per cent when *downstream-condensation* was avoided. It is therefore apparent from these observations, as well as from the theoretical considerations mentioned previously, that the location of the vent is critical. It is a corollary that effective venting is difficult or impossible in systems in which the pattern of steam flow is complex, as in many radiator systems. Generally, however, the type of equipment in which corrosion difficulties most frequently occur is of a design most favorable for the application of venting.

TABLE B—EFFECT OF *Downstream-Condensation* ON ELIMINATION OF CARBON DIOXIDE BY BLEEDING

PRESSURE, PSI	PER CENT BLED	DISTANCE CONDENSATE LEVEL BELOW BLEEDING POINT-INCHES	PER CENT ELIMINATION OF CO_2
40	1.1	19.5	49
40	1.0	12.5	54
40	1.0	4.6	67
40	1.1	1.0	92

In much of the discussion of the effect of carbon dioxide in steam heating systems, it seems not to have been generally appreciated that under some circumstances it is possible to form condensate at certain points in a system containing a much higher proportion of carbon dioxide than is present in the original steam. The author refers to one such case—a one-pipe system operated for a long time without interruption. A very high concentration of carbon dioxide may occur in the radiator because of the stripping of the carbon dioxide from the condensate of the return line and its return to the radiator by the rising steam. Instances have been noted in which the nipples joining radiator sections were so severely corroded that it was necessary to replace them. This is not commonly experienced in one-pipe systems, because in most

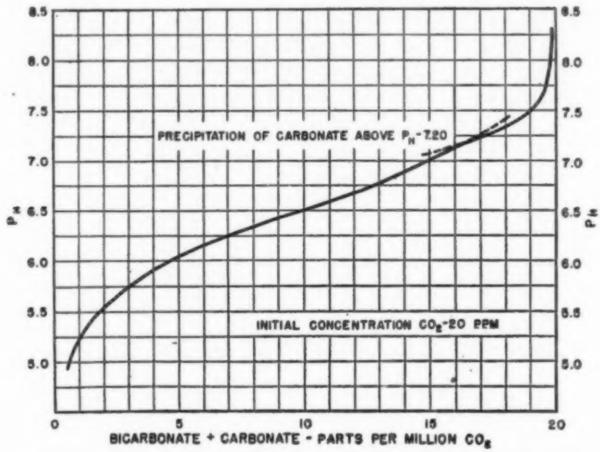

FIG. C. NEUTRALIZATION OF CARBONIC ACID BY IRON. INITIAL CONCENTRATION OF CO_2, 20 PPM

instances the controls operate to interrupt the supply at intervals too short to allow the collection of excessive amounts of carbon dioxide in the radiator.

Another location in which unusually high concentrations of carbon dioxide in condensate may occur is in return lines of inactive equipment connected to main return lines operating under some pressure. Residual vapor of high carbon dioxide content flowing back into these *pockets* may give condensates with concentrations of carbon dioxide as high as several hundred parts per million. The possibility of such high concentration of carbon dioxide is of particular concern because it is believed that there may be more than a proportional increase in corrosion with increasing carbon dioxide. The reason is that when iron is attacked by high concentrations of carbon dioxide the buffering action of iron corrosion product is not fully effective, because of the precipitation of iron carbonate. It is estimated that when carbon dioxide solution reacts with iron in the absence of oxygen at about 75-80 F, with concentrations of either 20 or 100 ppm, a pH of 6 is reached when about 24 per cent of the carbon dioxide has reacted with iron. On the other hand, with 1000 ppm of carbon dioxide in solution, it is estimated that a pH of 6 will not be reached until nearly 75 per cent of the carbon dioxide has reacted with iron. The data are not available for similar calculations for the higher temperatures corresponding to condensate in the return lines. It is probable that a precipitation of ferrous carbonate would likewise occur

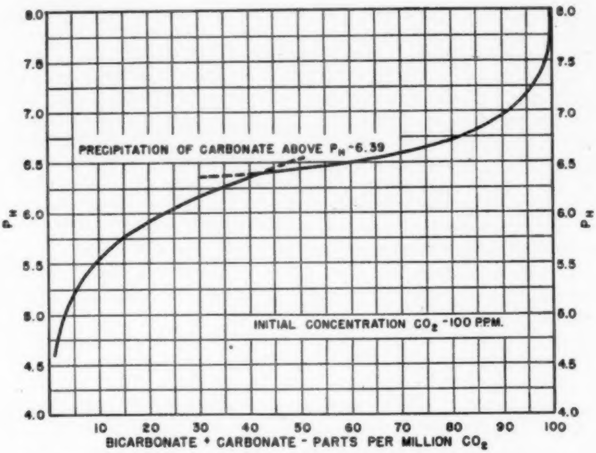

FIG. D. NEUTRALIZATION OF CARBONIC ACID BY IRON. INITIAL CONCENTRATION OF CO_2, 100 PPM

with higher concentration of carbon dioxide, although the quantitative relationships would be changed. The calculated curves for change in pH on continued action of carbon dioxide solution of the three different concentrations at about 75 F are shown in Figs. C, D and E. The calculations are based on available information, including data on the dissociation of carbonic acid and the solubility of ferrous carbonate.

There are one or two points in the data offered in the papers presented for which alternate interpretations might be considered. Thus it is not clear that, in the pres-

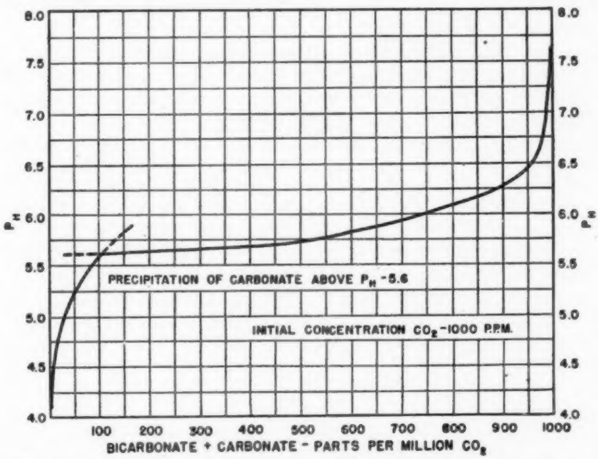

FIG. E. NEUTRALIZATION OF CARBONIC ACID BY IRON. INITIAL CONCENTRATION OF CO_2, 1000 PPM

ence of steam, sufficient undercooling would occur to significantly affect the entry of carbon dioxide into the condensate in the experimental water heaters, as suggested by the authors. In any case the concentration of carbon dioxide in the condensate is from five to fifty times that which would be in equilibrium with the vented steam. This must mean that much of the condensate has traveled through regions containing a higher partial pressure of carbon dioxide than is present at the point of purging. This factor is recognized by the authors and is perhaps alone sufficient to account for the variations noted without assuming any effects from *undercooling*.

The loss of CO_2 in passing through the meter noted in the Appendix of Mr. Collins' paper may be primarily a degassing effect due to the sweeping action of air. It does not seem probable that iron corrosion products would substantially affect the solubility of CO_2, except insofar as it is combined as iron bicarbonate.

FERDINAND JEHLE, Indianapolis, Ind.: I did not notice just how Dr. Guernsey controlled the point where he vented. It undoubtedly was on the second slide, but I think that I missed it.

P. H. WEITZEL, Dayton, Ohio: What is the generally accepted source of the gas? Is it dirty water? Is there any other way of solving the problem besides attempting to vent in some complicated manner?

DR. GUERNSEY: In the experimental apparatus (Fig. B) we controlled the level of the concentrate, observing it in a gage glass and then controlled it manually during the experiment simply by manipulating the valve F. The vertical tube was the line through which we vented steam. Then by manipulating the valve to maintain the level of the condensate a very short distance under that vent point, we could be certain that we were venting from practically the end of the path of travel of the steam. Of course, likewise we could open the valve in such a way that the condensate level could be carried at any desired lower point.

As I explained for that particular observation, we removed insulation from this tube to permit a small amount of condensation beyond this point. We do not know how much as we did not measure the amount. But we assumed that as compared with the cooling by the condensation of the water-cooled tube some condensation occurred beyond this point. Yet it was found that about 19 in. of this three-eighths inch tube was sufficient to reduce the venting from about 92 per cent to 49 per cent.

MR. JEHLE: What is part H in Fig. B?

DR. GUERNSEY: That merely indicates the means we used for sampling this condensate. We were operating at pressures above atmospheric in all the experiments. As we wanted to take the condensate out and analyze it, we used a cooling coil H through which we withdrew the condensate into our sampling bottles.

MR. COLLINS: To answer the question as to the source of CO_2 in the steam: The principal source of CO_2 in steam is the carbonate salts (usually calcium or magnesium bicarbonate) that are present in the raw water fed to the boiler. Under the conditions of operation these salts are decomposed with the liberation of CO_2.

D. M. HUMMEL, New Haven, Conn.: I would like to ask if CO_2 corrosion has any distinctive symptoms. One sees two kinds, for instance, grooving on the bottom of the pipe, and holes on the top.

MR. COLLINS: In general the pattern of oxygen corrosion is a pitted surface over which there is an accumulation of insoluble deposits. Carbon dioxide corrosion is typified by clean, evenly thinned surfaces. In partially filled lines, there is usually a grooving along the bottom.

1267

No. 1267

SUMMER WEATHER DATA AND SOL-AIR TEMPERATURE—STUDY OF DATA FOR NEW YORK CITY

By C. O. Mackey * and E. B. Watson,** Ithaca, N. Y.

THIS IS a discussion of a method of analysis of available summer weather data in order to obtain information of some value to engineers interested in the effect of solar heat upon cooling load and in other related problems. The intensity of the total radiation received from sun and sky on a horizontal surface is continuously recorded at several stations in the United States, maintained by the U. S. Weather Bureau and also at a number of cooperating stations. Hourly readings of the temperature of the outdoor air are also recorded. The intensity of direct solar radiation received in a plane perpendicular to the sun's rays is measured at four of these stations. Monthly summaries of these data appear in the Monthly Weather Review.

The sol-air temperature seems to be the most logical combination of these readings in analyzing problems in heat transfer. As explained in an earlier paper,† the *sol-air temperature* is the temperature of the outdoor air which, in contact with the shaded surface of any building material that does not directly transmit solar radiation, would give the same rate of heat transfer and the same temperature distribution through that material as exists with the actual outdoor air temperature and solar radiation incident upon the sunlit surface.

For either steady or unsteady flow of heat, the instantaneous rate of heat entry into the outside surface of a sunlit building material, which does not directly transmit solar radiation, is:

$$\left[\frac{q}{A}\right]_L = bI + h\,(t_a - t_L)\ \text{Btu/hr ft}^2,$$

where

b = absorptivity of the surface for solar radiation;
I = intensity of incident solar radiation, Btu/hr ft^2;
h = film coefficient of heat transfer between outdoor air and surface of the material Btu/hr ft^{2}F;
t_a = temperature of outdoor air, F;
t_L = temperature of surface of material, F.

Note that:

$$bI + h(t_a - t_L) = h\left[\frac{bI}{h} + t_a - t_L\right]$$

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** Assistant Professor of Engineering Materials, Cornell University.

† Summer Comfort Factors as Influenced by the Thermal Properties of Building Materials, by C. O. Mackey and L. T. Wright, Jr. (A.S.H.V.E. Transactions, Vol. 49, 1943, p. 148.)

Presented at the 51st Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, Mass., January, 1945.

Let the sol-air temperature be defined as:

$$t_e = t_a + \frac{bI}{h}$$

Then, the instantaneous rate of heat entry into the material at the outside surface may also be expressed as

$$\left[\frac{q}{A}\right]_L = h(t_e - t_L)$$

The sol-air temperature concept is new but extremely useful, because the effects of air temperature and solar radiation upon the rate of heat transfer and the temperature distribution through the sunlit material are combined. In other words, an increment in one degree in the modulus, $\frac{bI}{h}$, which has the dimensions of temperature, has precisely the same effect upon the temperature

TABLE 1—OUTDOOR AIR TEMPERATURE DATA—NEW YORK CITY

(Maximum hourly outdoor air temperature in the shade, and hourly outdoor air temperature equalled or exceeded on no more than 5 per cent of the days during a given month in the period from 1932 through 1941)

Time of Day (E.S.T.)	Temperature of the Outdoor Air in the Shade, deg F							
	Maximum				Equalled or Exceeded Only 5 Per Cent of the Total Hours, 1932–1941			
	June	July	August	September	June	July	August	September
1 AM	80	84	85	79	76	78	77	73
2	78	82	84	78	75	77	76	72
3	77	81	83	77	74	77	75	72
4	77	81	82	77	73	76	75	71
5	77	80	81	75	74	76	74	71
6	81	82	82	75	75	76	75	71
7	84	86	84	78	77	80	78	72
8	86	90	86	80	80	82	80	74
9	90	92	91	83	82	86	83	77
10	91	95	93	86	84	88	86	80
11	94	98	95	89	87	90	88	83
12	96	100	96	90	88	92	90	85
1 PM	97	102	98	93	90	93	91	87
2	98	104	100	96	91	94	91	87
3	99	105	99	96	90	94	91	88
4	100	106	96	92	89	94	91	86
5	97	105	97	90	89	93	90	85
6	94	103	96	87	87	90	88	82
7	92	94	94	86	84	88	85	79
8	88	93	93	84	82	85	83	77
9	86	92	91	83	80	83	81	76
10	85	88	89	82	79	82	80	75
11	83	85	87	80	78	81	78	74
12	81	84	86	79	76	79	77	73
24-hour average of above	88.0	92.1	90.3	84.0	81.7	84.8	82.6	77.9

TABLE 2—INTENSITY OF TOTAL SOLAR AND SKY RADIATION ON A HORIZONTAL SURFACE —NEW YORK CITY

(Maximum hourly intensity, and hourly intensity equalled or exceeded on no more than 5 per cent of the days during a given month in the period from 1932 through 1941; all times are Eastern Standard)

TIME OF DAY	INTENSITY OF TOTAL SOLAR AND SKY RADIATION ON HORIZONTAL PLANE, BTU/HR FT²							
	MAXIMUM				EQUALLED OR EXCEEDED ONLY 5 PER CENT OF THE TOTAL HOURS, 1932–1941			
	June	July	August	September	June	July	August	September
6 AM	45	51	28	...	30	28	15	...
7	127	127	84	52	91	80	59	31
8	206	160	146	154	153	144	126	87
9	279	216	211	183	205	196	180	151
10	318	268	261	224	249	236	210	197
11	347	295	288	252	287	260	244	224
12	362	305	301	270	302	278	266	241
1 PM	360	300	305	279	287	278	264	239
2	343	289	292	268	260	267	241	221
3	326	258	256	226	280	225	216	186
4	258	215	206	183	215	189	171	150
5	187	166	142	118	163	137	123	91
6	112	108	84	66	85	82	65	37
7	38	44	31	...	35	31	19	...

of the *inside* surface, and the rate of heat transfer at that surface contributing to the cooling load, as would an increment of one degree in the temperature of the outdoor air.

A more complete discussion of sol-air temperature, with numerical examples, is given in Appendix—*A* for materials which do not directly transmit solar radiation, and in Appendix—*E* for glass

Hourly readings of the dry-bulb temperature of the outdoor air and the total solar and sky radiation incident upon a *horizontal* surface for each hour of each day during the months of June, July, August, and September for the ten-year period from 1932 through 1941 were obtained from data reported by the New York Meteorological Observatory located in Central Park at a north latitude of 40°46′, a west longitude of 73°58′, and at an elevation of 180 ft above sea level.‡ Considerable atmospheric contamination reduces the intensity of solar radiation received at this station, but Dr. I. F. Hand of the U. S. Weather Bureau believes that *values representative of average large city conditions* are obtained.

Table 1 gives temperature data; the maximum temperature at each hour during June, for example, is the maximum of 300 recorded temperatures; the temperature listed as equalled or exceeded by that at only 5 per cent of the total hours would be that temperature equalled or exceeded in June, for example, by 15 of the 300 readings for that hour.

Table 2 gives solar radiation data; the maximum intensity of total solar and

‡ Tables presented here are taken from a thesis by E. B. Watson for the degree of Master of Science in Engineering at Cornell University.

sky radiation received on a horizontal surface is listed for each hour during each month of the ten-year period. Also the hourly intensity equalled or exceeded on no more than 5 per cent of the days at a given hour during each month of the ten-year period is given.

Table 3 gives the maximum sol-air temperature for the months of June, July, August and September for a horizontal surface with three solar absorptivities 1.0, 0.7 and 0.4. All sol-air temperatures are based upon Equation 1 of this report (Appendix—*A*). Careful observers may note what appears to be a discrepancy in Tables 1, 2 and 3. For example, *if* the maximum air temperature and the maximum solar radiation had occurred *simultaneously* at 12 noon in June, the maximum sol-air temperature for the horizontal surface with solar absorptivity, *b*, of 1.0 would have been $\left\{96 + \frac{362}{4}\right\}$ or 186.5 F; actually, the maximum sol-air temperature for this hour and month was observed to be only 162 F. The reason for this is the observed fact that *the maximum air temperatures and the maximum solar radiation do not, in general, occur simultaneously.* It is not correct, therefore, to combine maximum

TABLE 3—MAXIMUM SOL-AIR TEMPERATURE FOR HORIZONTAL SURFACE IN NEW YORK CITY

(*Maximum values for 10-year period, 1932–1941*)

TIME OF DAY (E.S.T.) (SOLAR ABSORPTIVITY)	SOL-AIR TEMPERATURE, DEG F											
	June			July			August			September		
	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)
1 AM...	80	80	80	84	84	84	85	85	85	79	79	79
2 ...	78	78	78	82	82	82	84	84	84	78	78	78
3 ...	77	77	77	81	81	81	83	83	83	77	77	77
4 ...	77	77	77	80	80	80	82	82	82	77	77	77
5 ...	77	77	77	82	82	82	81	81	81	75	75	75
6 ...	86	85	85	86	85	83	86	85	84	75	75	75
7 ...	102	95	90	104	98	92	98	94	90	89	85	81
8 ...	125	111	96	118	108	100	113	103	95	107	95	86
9 ...	141	123	106	139	124	108	130	117	105	119	106	95
10 ...	149	127	111	148	132	116	145	128	111	134	118	103
11 ...	155	135	115	157	139	122	150	132	116	145	126	108
12 ...	162	140	118	162	143	125	160	141	121	150	131	112
1 PM...	163	145	122	170	146	126	161	141	121	151	132	114
2 ...	164	142	122	163	145	127	161	141	121	150	137	115
3 ...	155	137	120	169	150	131	153	135	118	141	125	111
4 ...	143	129	117	144	133	121	140	127	113	128	117	106
5 ...	131	121	110	131	123	116	125	115	106	111	104	98
6 ...	115	108	102	113	110	107	109	104	99	95	93	90
7 ...	100	98	95	100	98	96	94	96	95	86	86	86
8 ...	88	88	88	93	93	93	93	93	93	84	84	84
9 ...	86	86	86	92	92	92	91	91	91	83	83	83
10 ...	85	85	85	88	88	88	89	89	89	82	82	82
11 ...	83	83	83	85	85	85	87	87	87	80	80	80
12 ...	81	81	81	84	84	84	86	86	86	79	79	79
24-hour average of above.	112.6	104.5	96.7	114.8	107.7	100.9	111.9	105.0	98.2	103.1	96.8	90.6

values of the two effects and the need for some concept like the sol-air temperature becomes apparent. Another apparent discrepancy may be found if one assumes that the difference between the sol-air temperatures for $b = 1$ and $b = 0.7$ should be the same as the difference for $b = 0.7$ and $b = 0.4$; this need not be the case, however, for the temperature of the air becomes more important when the solar absorptivity decreases. In other words, the maximum sol-air temperature for $b = 1$ might occur on the day when the solar radiation is a maximum, while for $b = 0.4$ the sol-air temperature may be a maximum on the day when the temperature of the air is a maximum. This latter effect is small and for most practical purposes the difference between the sol-air temperature for any solar absorptivity and the sol-air tem-

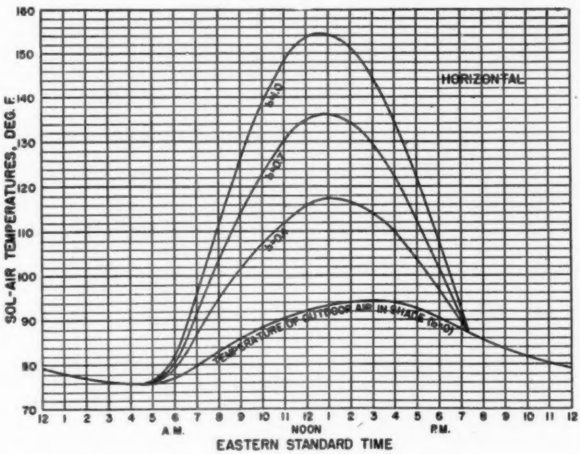

FIG. 1. DESIGN TEMPERATURES FOR HORIZONTAL SURFACE IN NEW YORK, N. Y.

perature for zero absorptivity (air temperature in the shade) may be assumed to be directly proportional to that solar absorptivity.

Since the 24-hour average of the sol-air temperature is greater for July than for any other month, the sol-air temperature at each hour in July which is equalled or exceeded at that hour only 16 times in 310 observations in July (ten-year period) has been chosen as the design sol-air temperature of the outdoor air. These temperatures are plotted in Fig. 1 for a horizontal surface and the result is a design curve for New York City which is recommended for use in calculations of heat transfer through horizontal surfaces in that locality. Similar curves must be found for other localities before any general design curves may be drawn.

The design curves include the air temperature in the shade (solar absorptivity of zero) and three curves of sol-air temperatures for solar absorptivities of 1.0, 0.7 and 0.4. Although data on solar absorptivity are not very complete, it is recommended that a solar absorptivity of 1.0 be used for very dark colors, 0.7 for medium colors and 0.4 for very light colors; the air tempera-

ture curve should be used for all surfaces completely shaded from direct solar and diffuse sky radiation.

VERTICAL SURFACES

It would be desirable to have the sol-air temperature of the outdoor air for vertical surfaces of various orientations in order that the flow of heat through

TABLE 4—SOL-AIR TEMPERATURE FOR HORIZONTAL SURFACE IN NEW YORK CITY EQUALLED OR EXCEEDED AT ANY HOUR ONLY 5 PER CENT OF THE TOTAL HOURS IN A TEN-YEAR PERIOD, 1932–1941

TIME OF DAY (E.S.T.) (SOLAR ABSORPTIVITY	SOL-AIR TEMPERATURE, DEG F											
	June			July			August			September		
	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)
1 AM...	76	76	76	78	78	78	77	77	77	73	73	73
2 ...	75	75	75	77	77	77	76	76	76	72	72	72
3 ...	74	74	74	77	77	77	75	75	75	72	72	72
4 ...	73	73	73	76	76	76	75	75	75	71	71	71
5 ...	74	74	74	76	76	76	74	74	74	71	71	71
6 ...	79	78	76	81	79	78	78	77	77	71	71	71
7 ...	93	88	83	96	92	87	89	85	81	78	77	75
8 ...	108	99	91	110	102	93	106	98	90	91	84	82
9 ...	123	111	99	127	114	101	122	111	97	106	97	88
10 ...	137	121	105	137	122	108	131	117	103	121	108	94
11 ...	143	127	109	148	130	112	141	124	108	131	116	103
12 ...	149	131	112	155	135	116	148	129	113	136	121	106
1 PM...	148	130	116	154	136	118	150	132	114	141	124	108
2 ...	150	132	113	152	134	116	146	130	113	136	121	106
3 ...	141	125	110	144	129	113	140	125	110	128	115	102
4 ...	132	119	106	136	123	110	129	118	106	118	107	97
5 ...	121	111	101	121	112	104	116	108	100	101	95	91
6 ...	104	99	94	106	101	96	101	97	91	87	85	83
7 ...	92	89	87	93	91	90	88	87	86	79	79	79
8 ...	82	82	82	85	85	85	83	83	83	77	77	77
9 ...	80	80	80	83	83	83	81	81	81	76	76	76
10 ...	79	79	79	82	82	82	80	80	80	75	75	75
11 ...	78	78	78	81	81	81	78	78	78	74	74	74
12 ...	76	76	76	79	79	79	77	77	77	73	73	73
24-hour average of above.	103.6	97.0	90.3	106.4	99.8	93.2	103.0	96.4	90.2	94.1	88.9	84.2

such surfaces might be found. The Weather Bureau has made very few observations of the intensity of solar radiation received on vertical surfaces. If it were not for the fact that part of the total observed radiation received upon the horizontal surface is diffuse or sky radiation, it would be relatively simple to calculate the intensity on a vertical surface from the observations on the horizontal surface. (For direct or beamed solar radiation, the ratio of the intensity on a vertical surface to the intensity on a horizontal surface may be found as explained in Appendix—*B*).

As solar radiation passes through the earth's atmosphere, part is turned aside from the direct beam and scattered in practically all directions with no

appreciable change in wave length. A considerable portion of the total radiation received on a horizontal surface is in the form of this diffuse radiation from the sky. On cloudy days all radiation is diffuse, while on clear days the ratio of diffuse to direct radiation varies with the solar altitude and with the amount of dust, water vapor and other material in the atmosphere. This ratio is large for low solar altitudes, cloudy days and large amounts of smoke

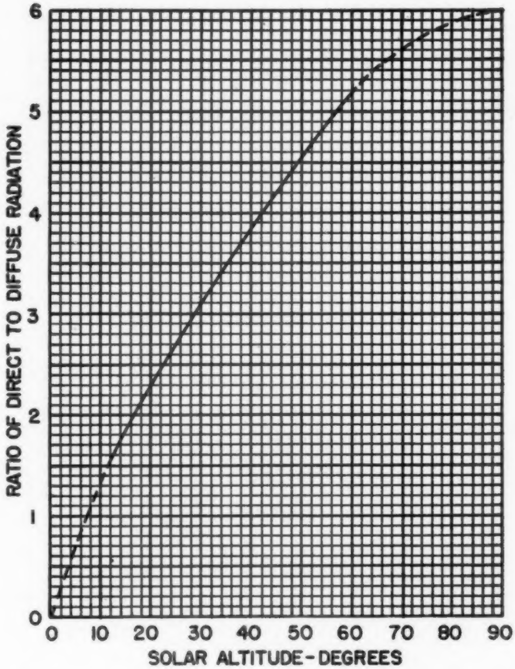

FIG. 2. RATIO OF DIRECT TO DIFFUSE RADIATION ON HORIZONTAL SURFACES; SUMMER DATA; EASTERN STATES

and dust in the atmosphere. On clear days, in cities near the times of sunrise and sunset, the diffuse radiation generally amounts to more than one-half of the total received on a horizontal surface, while on high mountains the diffuse radiation is almost negligible except with very low sun.

From a large number of measurements of diffuse radiation, Dr. I. F. Hand gives the following table [1] for the ratio of direct solar radiation on a horizontal surface to diffuse sky radiation on that surface during cloudless days in Washington, D. C. as indicated in the tabulation.

[1] Monthly Weather Review, 1937, page 437.

Solar altitude, H	60°	30°	11.3°
Ratio of direct to diffuse radiation, summer	5.2	3.1	1.5
Ratio of direct to diffuse radiation, winter	8.1	5.0	1.7

The summer results are shown plotted in Fig. 2, in the form of the ratio of direct to diffuse radiation received on a horizontal surface as a function of solar altitude. The resulting curve is typical for eastern states.

The point of greatest uncertainty in the extension of data obtained from radiation incident upon horizontal surfaces is the question of how the diffuse sky radiation on a vertical surface compares with the diffuse sky radiation on a horizontal surface. What fraction of this diffuse sky radiation on a horizontal surface is incident upon vertical surfaces of different orientations at different times of the day? Only more complete radiation data would answer this question. In this report, for simplicity, it has been assumed that the diffuse sky radiation incident on a vertical surface of any orientation is one-half of that incident upon a horizontal surface at the same time of day.

TABLE 5—SOL-AIR TEMPERATURES FOR VERTICAL SURFACES IN NEW YORK CITY EQUALLED OR EXCEEDED AT ANY HOUR IN JULY ONLY 5 PER CENT OF THE TOTAL HOURS IN A TEN-YEAR PERIOD, 1932–1941

(*Calculated values based upon assumptions stated in the body of the paper.*)

TIME OF DAY (E.S.T.)	SOL-AIR TEMPERATURE, DEG F											
	SURFACE FACING											
	North			East			South			West		
(SOLAR ABSORPTIVITY)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)
1 AM...	78	78	78	78	78	78	78	78	78	78	78	78
2 ...	77	77	77	77	77	77	77	77	77	77	77	77
3 ...	77	77	77	77	77	77	77	77	77	77	77	77
4 ...	76	76	76	76	76	76	76	76	76	76	76	76
5 ...	76	76	76	76	76	76	76	76	76	76	76	76
6 ...	80	79	77	89	85	81	77	77	76	77	77	76
7 ...	85	83	82	106	98	91	82	82	81	82	82	81
8 ...	85	84	83	114	105	95	86	85	83	85	84	83
9 ...	90	89	88	120	109	99	97	94	90	90	89	88
10 ...	92	91	90	114	106	98	104	99	94	92	91	90
11 ...	94	93	92	106	101	97	111	105	98	94	93	92
12 ...	97	95	93	97	95	94	115	108	101	98	96	94
1 PM...	98	96	95	98	96	95	117	110	103	108	104	99
2 ...	99	97	96	99	97	96	112	107	101	126	116	107
3 ...	99	97	96	99	97	96	102	99	97	136	123	111
4 ...	99	97	96	99	97	96	99	98	96	145	130	114
5 ...	103	100	97	97	96	95	97	96	95	140	126	112
6 ...	106	101	96	93	92	91	93	92	91	134	120	107
7 ...	102	98	94	90	89	89	90	89	89	115	107	99
8 ...	85	85	85	85	85	85	85	85	85	85	85	85
9 ...	83	83	83	83	83	83	83	83	83	83	83	83
10 ...	82	82	82	82	82	82	82	82	82	82	82	82
11 ...	81	81	81	81	81	81	81	81	81	81	81	81
12 ...	79	79	79	79	79	79	79	79	79	79	79	79
24-hour average of above.	88.5	87.3	86.2	92.3	89.9	87.8	90.7	89.0	87.1	96.5	93.0	89.5

Based upon these assumptions, the sol-air temperatures were calculated for the month of July and for vertical surfaces of various orientations and are shown in Table 5. These are the sol-air temperatures that were equalled or exceeded on no more than 5 per cent of the days during that month in the ten-year period from 1932 to 1941. These values are not as reliable as those

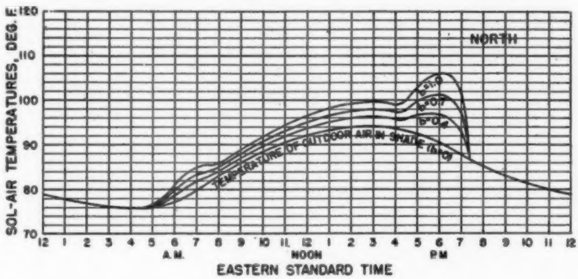

FIG. 3. DESIGN TEMPERATURES FOR SURFACE FACING NORTH IN NEW YORK, N. Y.

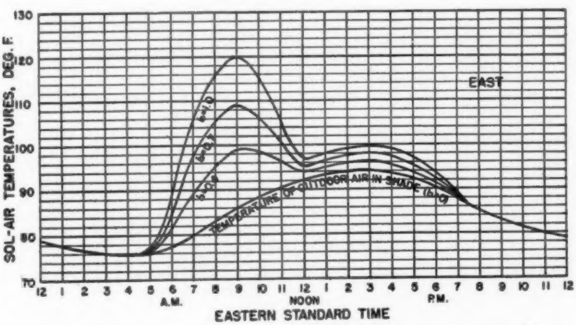

FIG. 4. DESIGN TEMPERATURES FOR SURFACE FACING EAST IN NEW YORK, N. Y.

for the horizontal surface, because they are based upon the assumed breakdown of total radiation into direct and diffuse and a further assumed ratio of diffuse radiation on the vertical surfaces to diffuse radiation on a horizontal surface. However, the curve of Fig. 2 is believed to be fairly reliable, and the effect of the assumption concerning the diffuse radiation incident upon the vertical surface is relatively unimportant in the final result as is shown in Appendix—*C*.

The recommended design curves for sol-air temperature for New York City are shown for vertical surfaces with different values of solar absorptivity facing north (Fig. 3), facing east (Fig. 4), facing south (Fig. 5) and facing west (Fig. 6).

EQUATIONS FOR SOL-AIR TEMPERATURE

For the few who may be interested, equations for the recommended design sol-air temperatures for New York City are presented in the form of trigo-

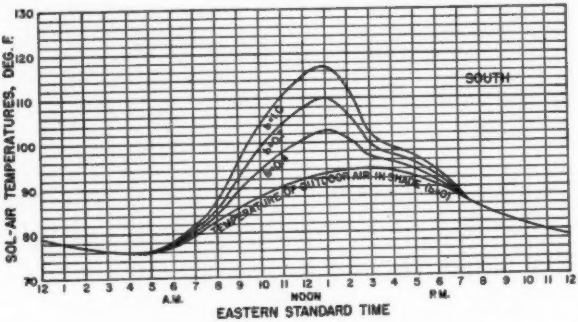

FIG. 5. DESIGN TEMPERATURES FOR SURFACE FACING SOUTH IN NEW YORK, N. Y.

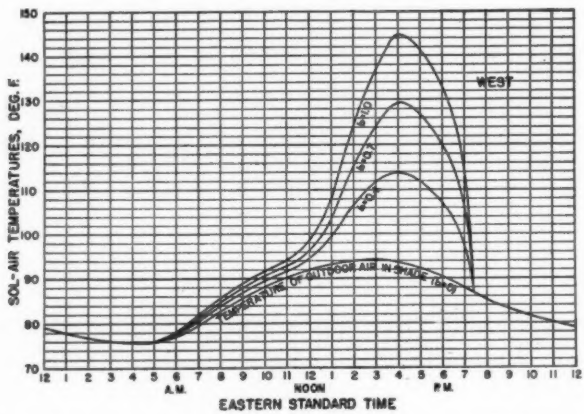

FIG. 6. DESIGN TEMPERATURES FOR SURFACE FACING WEST IN NEW YORK, N. Y.

nometric series. In all of these equations, θ represents the time in hours after noon (E.S.T.).

Design temperature of outdoor air in shade:

$$t_a = 84.8 + 9.0 \cos(15\theta - 42) + 1.1 \cos(30\theta - 28) + 0.7 \cos(45\theta - 228) + 0.3 \cos(60\theta - 225)$$

Design sol-air temperature of outdoor air for horizontal surfaces:

Solar absorptivity of 1.0,

$$t_e = 106.4 + 39.7 \cos (15\theta - 15) + 11.5 \cos (30\theta - 17) + 2.0 \cos (45\theta - 210) + 1.4 \cos (60\theta - 217)$$

Design sol-air temperature of outdoor air for vertical surfaces facing *North:*

Solar absorptivity of 1.0,

$$t_e = 88.5 + 12.9 \cos (15\theta - 44) + 1.8 \cos (30\theta - 165) + 2.6 \cos (45\theta - 253) + 2.1 \cos (60\theta - 12)$$

Design sol-air temperature for vertical surfaces facing *East:*

Solar absorptivity of 1.0,

$$t_e = 92.3 + 15.5 \cos (15\theta - 356) + 6.9 \cos (30\theta - 262) + 8.3 \cos (45\theta - 209) + 2.6 \cos (60\theta - 154)$$

Design sol-air temperature for vertical surfaces facing *South:*

Solar absorptivity of 1.0,

$$t_e = 90.7 + 17.6 \cos (15\theta - 22) + 6.3 \cos (30\theta - 5) + 2.3 \cos (45\theta - 284) + 0.9 \cos (60\theta - 5)$$

Design sol-air temperature for vertical surfaces facing *West:*

Solar absorptivity of 1.0,

$$t_e = 96.5 + 28.1 \cos (15\theta - 54) + 12.6 \cos (30\theta - 125) + 8.0 \cos (45\theta - 203) + 2.2 \cos (60\theta - 281)$$

SOL-AIR TEMPERATURE FOR GLASS

The sol-air temperatures given in this paper are to be used only for estimating the rate of heat flow from the indoor surfaces of materials which do not directly transmit incident solar radiation. Part of the total radiation incident upon glass is directly transmitted, part is reflected and part is absorbed. If the sol-air temperature concept is to be used for predicting the heat flow through glass, this temperature must be increased to allow for the direct transmission of some of the incident radiation. Exact application requires a knowledge of the transmissivity and absorptivity of glass as a function of the angle of incidence of the sun's rays. This point is discussed further in Appendix—*E*. The simplest procedure in obtaining design information concerning the contribution to the cooling load due to heat transfer through glass would probably be to multiply the values of total radiation received on a horizontal surface, as given in Table 2, by an appropriate factor at each hour; this factor would include the combined effects of glass orientation and the effect of angle of incidence upon absorptivity and transmissivity.

APPENDIX

METHODS AND EQUATIONS

A. Sol-Air Temperature

For the surface of a material which does not directly transmit solar radiation, the sol-air temperature may be found from observed values of the air temperature and the total solar and sky radiation incident upon that surface as follows:

$$t_e = t_a + \frac{bI}{h} \quad \ldots \quad (1)$$

where

t_e = the sol-air temperature, F;
t_a = the dry-bulb temperature of the outdoor air in the shade, F;
I = the intensity of total solar and sky radiation incident upon the surface, Btu/hr ft^2;
b = the absorptivity of the surface for the incident total radiation;
h = the film coefficient of heat transfer between surface and air, Btu/hr ft^2 F. (Actually, this coefficient depends upon wind velocity, but it was assumed to be a constant and equal to 4 in this study.)

Example: Observed dry-bulb temperature of outdoor air in the shade is $t_a = 78$ F; observed total radiation incident upon *horizontal* surface is $I = 200$ Btu/hr ft^2. For a surface absorptivity of 0.7, the sol-air temperature for the *horizontal* surface is:

$$t_e = 78 + \frac{0.7\ (200)}{4} = 113\ \text{F}$$

In other words, air at a dry-bulb temperature of 113 F in contact with the completely *shaded* surface would give the same temperature distribution and rate of heat flow through the building material as exists with an air temperature of 78 F, incident total radiation of 200 Btu/hr ft^2 and a surface absorptivity of 0.7. This concept permits combining the effects of air temperature and solar radiation.

B. Trigonometric Relations

The intensity of *direct* solar radiation on a horizontal surface is substantially equal to the product of the intensity of direct solar radiation received on a plane perpendicular to the sun's rays and the cosine of the angle between the sun's rays and the normal to the horizontal, or

$$I_H = K_H\, I_o \qquad (2)$$

where

I_H = the intensity of *direct* solar radiation received on a horizontal surface, Btu/hr ft^2;
I_o = the intensity of *direct* solar radiation received on a plane perpendicular to the sun's rays, Btu/hr ft^2.

$$K_H = \cos (90 - H) \qquad (3)$$

or

$$K_H = \sin D \sin L + \cos D \cos L \cos (360 - \theta) \qquad (4)$$

where

H = the solar altitude, degrees;
D = the declination of the sun, degrees;
L = the latitude of the surface, degrees;
θ = the hour angle of the sun measured west from the zenith, degrees.

The intensity of *direct* solar radiation on a vertical surface is:

$$I_v = K_v\, I_o \qquad (5)$$

where

I_v = the intensity of *direct* solar radiation received on a vertical surface, Btu/hr ft^2;

$$K_v = \cos H \cos B \qquad (6)$$

and

$$B = A_v - 90 - A_s \qquad (7)$$

where

A_v = the azimuth of the trace of the vertical surface, measured west from south, degrees;
A_s = the azimuth of the sun measured west from south, degrees, (If cos B is zero or negative, the vertical surface is receiving no *direct* solar radiation.)

The azimuth of the sun may be found from the following relation:

$$\sin (A_s - 180) = \frac{\cos D \sin (360 - \theta)}{\cos H} \qquad (8)$$

Table A—Solar Altitudes at Different Times for New York City in Mid-July

Time (E.S.T.)	Solar Altitude North Latitude of 40° 46′ West Longitude of 73° 58′ and Solar Declination of 21° 36′
4:36 AM	0 (sunrise)
5	4° 2′
6	14° 40′
7	25° 47′
8	37° 7′
9	48° 21′
10	58° 56′
11	67° 29′
12 noon	70° 49′
1 PM	66° 31′
2	57° 33′
3	46° 50′
4	35° 33′
5	24° 14′
6	13° 10′
7	2° 38′
7:16	0 (sunset)

If the *direct* solar radiation received on a horizontal surface is known, the *direct* solar radiation that is then incident upon a vertical surface of any orientation may be computed.

Tables A and B used in the calculations are offered here because they may be of some interest in similar calculations. Table A gives the solar altitudes at different times for a north latitude of 40°46′ and a west longitude of 73°58′ (New York) and a solar declination of 21°36′ (mid-July).

Table B—Ratio of Direct Solar Radiation on Vertical to Radiation on Horizontal Surface for New York City in Mid-July

Time (E.S.T.)	Ratio: Direct Solar Radiation on Vertical Surface to Direct Solar Radiation on Horizontal Surface							
	Vertical Surface Facing							
	N	NE	E	SE	S	SW	W	NW
5 AM	6.044	13.335	12.814	4.787	...	...	...	...
6	1.059	3.346	3.673	1.848	...	...	...	...
7	0.254	1.633	2.055	1.273	...	...	...	...
8	...	0.894	1.321	0.973	0.056	...	...	...
9	...	0.460	0.864	0.762	0.213	...	...	...
10	...	0.167	0.526	0.580	0.295	...	...	...
11	...	...	0.243	0.481	0.335	0.065	...	...
12 noon	...	...	...	0.238	0.349	0.259	0.018	...
1 PM	...	...	...	0.003	0.333	0.433	0.280	...
2	...	...	...	...	0.287	0.604	0.568	0.205
3	...	...	...	...	0.196	0.788	0.918	0.510
4	...	...	...	...	0.028	1.009	1.399	0.970
5	...	...	...	...	...	1.326	2.198	1.783
6	...	...	...	...	...	1.987	4.083	3.787
7	...	...	...	...	...	6.889	19.900	20.645

In all of this study, the small difference between mean solar time and apparent solar time was ignored. Since New York is east of the 75th meridian, the mean sun is 1°2′ ahead of the position indicated by Eastern Standard Time. Although the position of the *mean* sun may be corrected to the position of the *true* sun through the equation of time, this correction was ignored.

Table B gives the ratio of direct solar radiation on a vertical surface to direct solar radiation on a horizontal surface for a north latitude of 40°46′, a west longitude of 73°58′ and a solar declination of 21°36′ (New York in mid-July).

The assumption with the least data to authorize it and, fortunately, the one with the least effect upon the result, is that the intensity of diffuse sky radiation on a vertical surface, regardless of orientation and sun position, is one-half of the intensity of diffuse sky radiation on a horizontal surface at the same time. The steps in finding the diffuse radiation incident upon a vertical surface corresponding to the observed total radiation on a horizontal surface equalled or exceeded on no more than 5 per cent of the days during the month of July (Table 2) are shown in Table C. The solar altitude at each hour is calculated on the assumption of a solar declination of 21°36′ (mid-July).

TABLE C—DETERMINATION OF DIFFUSE RADIATION ON VERTICAL SURFACE FROM TOTAL SKY AND SKY RADIATION ON A HORIZONTAL SURFACE

Time (E.S.T.)	Total Solar and Sky Radiation on a Horizontal Surface, Btu/hr ft² (From Table 2)	Solar Altitude	Ratio of Direct to Diffuse Radiation on Horizontal Surface (From Fig. 2)	Diffuse Radiation Incident Upon Horizontal Surface Btu/hr ft²	Diffuse Radiation Incident Upon Vertical Surface (Assumed)
6 AM......	28	14° 40′	1.80	10.0	5.0
7	80	25° 47′	2.75	21.4	10.7
8	144	37° 7′	3.60	31.4	15.7
9	196	48° 21′	4.40	36.3	18.2
10	236	58° 56′	5.13	38.5	19.3
11	260	67° 29′	5.50	40.0	20.0
12	278	70° 49′	5.61	42.0	21.0
1 PM......	278	66° 31′	5.48	43.0	21.5
2	267	57° 33′	5.03	44.3	22.2
3	225	46° 50′	4.28	42.7	21.4
4	189	35° 33′	3.50	42.0	21.0
5	137	24° 14′	2.65	37.5	18.8
6	82	13° 10′	1.67	30.7	15.4
7	31	2° 38′	0.35	23.0	11.5

Some data for the observed intensities of diffuse sky radiation on vertical surfaces facing south, east and west were plotted by Houghten, Shore, Olson and Gunst in the A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 95. From these data, a suggested design curve for diffuse sky radiation on vertical surfaces was obtained (see Fig. C-1). This curve is compared with the data obtained in Table C. The diffuse sky radiation on a vertical surface depends upon the orientation of that surface with respect to the sun as well as the solar altitude and is not a constant fraction of the diffuse radiation on a horizontal surface at the same time. However, it is believed that the diffuse radiation assumed in preparing the sol-air temperature curves for vertical surfaces in this report is within about 10 Btu/hr ft² of the actual and the equivalent temperatures for vertical surfaces are not over 3 F in error, even for the case where the solar absorptivity is 1.0.

It should be noted that observed values of the total radiation incident upon a horizontal surface must always be broken down into direct and diffuse radiation before any attempt is made to *estimate* the total radiation incident upon a vertical surface and the trigonometric calculation will apply only to the *direct* radiation. For exam-

ple, assume the total solar radiation received on a horizontal surface at 6 p.m. in New York in mid-July is 82 Btu/hr ft² and that it is required to find the total radiation incident at that time upon a vertical wall facing west. The ratio of *direct* solar radiation on this vertical wall to the *direct* solar radiation on a horizontal surface at this time is 4.083. If one were to assume that this was also the ratio of *total* radiation, one might conclude that the total radiation incident upon the west wall would be 4.083 (82) or 323 Btu/hr ft². This is incorrect, however, because about 31 of the 82 Btu/hr ft² incident upon the horizontal is diffuse sky radiation; hence, the total radiation incident upon the west wall at this time is more nearly [4.083 (51) + 0.5 (31)] or 224 Btu/hr ft².

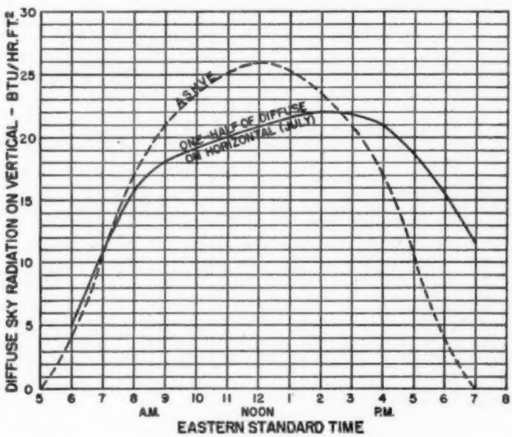

FIG. C-1. SUGGESTED DESIGN CURVE FOR DIFFUSE SKY RADIATION ON VERTICAL SURFACES

D. Sol-Air Temperature for Vertical Surfaces

The method used in obtaining the sol-air temperature for a vertical surface is illustrated by the following sample calculation:

At 9 a.m. (E.S.T.), the observed temperature of the outdoor air in the shade at the New York Meteorological Observatory is $t_a = 78$ F and the observed total solar and sky radiation incident upon the horizontal surface is $I = 200$ Btu/hr ft². The declination of the sun at this time is $D = 21°36'$. It is desired to find the sol-air temperature at this time for a vertical surface with an absorptivity of $b = 1.0$, facing *east.*

From Equation 4,

$$K_H = (\sin 21° 36')(\sin 40° 46') + (\cos 21° 36')(\cos 40° 46')(\cos 43° 58')$$
$$= 0.7472$$

Since $\sin H = 0.7472$, the solar altitude at this time is $H = 48°21'$.

From Fig. 2, the ratio of direct solar radiation to diffuse sky radiation for this solar altitude is 4.4. The diffuse sky radiation on the horizontal surface is 200/5.4 or

37 Btu/hr ft^2, while the direct solar radiation on the horizontal surface is 163 Btu/hr ft^2.

The azimuth of the sun at this time, A_s, may be found from Equation 8:

$$\sin (A_s - 180) = \frac{(\cos 21^\circ\, 36')(\sin 43^\circ\, 58')}{\cos 48^\circ\, 21'}$$
$$= 0.9712, \text{ or}$$
$$A_s = 283^\circ\, 47'$$

For a vertical surface facing *east*, the azimuth of the trace of the surface is $A_v = 360^\circ$, and from Equation 7,

$$B = 360^\circ - 90^\circ - 283^\circ\, 47' = -13^\circ\, 47'$$

From Equation 6,

$$K_v = (\cos 48^\circ\, 21')(\cos -13^\circ\, 47') = 0.6455$$

The intensity of *direct* solar radiation on the vertical surface facing east is:

$$I_v = \frac{0.6455}{0.7472}(163) = 140.8 \text{ Btu/hr ft}^2$$

The intensity of diffuse sky radiation on the vertical surface is assumed to be one-half of that on the horizontal surface or 18.5 Btu/hr ft^2.

The *total* solar and sky radiation incident on the vertical surface facing east is:

$$I = 140.8 + 18.5 = 159.3 \text{ Btu/hr ft}^2$$

For a solar absorptivity of $b = 1.0$, the sol-air temperature at this time for the vertical surface facing east is, from Equation 1,

$$t_e = 78 + \frac{159.3}{4} = 118 \text{ F}$$

E. Sol-Air Temperature for Glass

Assume that the fraction of incident total radiation which is transmitted directly through glass, the transmissivity, is designated by t; the fraction absorbed, absorptivity, by b; the fraction reflected, reflectivity, by r.

then,

$$t + b + r = 1$$

For the steady flow of heat through glass (no heat storage), it may be shown that the contribution to the cooling load per square foot of glass due to heat directly transmitted and heat conducted through the glass is:

$$\frac{q}{A} = \left(\frac{q}{A}\right)_t + \left(\frac{q}{A}\right)_c$$
$$= tI + U\left[(t_a - t_i) + \frac{bI}{h}\right] \quad \quad (9)$$

where

$\frac{q}{A}$ = the contribution to the cooling load, Btu/hr ft^2,

I = the total solar and sky radiation incident upon the outdoor surface of the glass, Btu/hr ft^2,

U = the overall coefficient of heat transfer of the glass, Btu/hr ft^2 F,

h = the film coefficient of heat transfer at the outside surface of the glass, Btu/hr ft^2 F,

t_a = the temperature of the outdoor air, F,

t_i = the temperature of the indoor air, Fahrenheit degrees.

Note that

$$\frac{q}{A} = U\left[\left(t_a + \frac{bI}{h} + \frac{tI}{U}\right) - t_i\right] \quad \text{(10a)}$$

$$= U\,(t_e - t_i) \quad \text{(10b)}$$

The sol-air temperature for glass is:

$$t_e = t_a + I\left(\frac{b}{h} + \frac{t}{U}\right) \quad \text{(11)}$$

Unfortunately, the absorptivity, b, and the transmissivity, t, are not constant but vary with the angle of incidence of the sun's rays; this variation would have to be known for an exact application of this principle.

As an illustration of the use of Equation 11, consider the following case:

Assume the intensity of the incident solar and sky radiation is 300 Btu/hr ft^2 on a surface at normal incidence to the sun's rays; assume $t_a = 90$ F, $h = 4$ Btu/hr ft^2 F and $U = 1.13$ Btu/hr ft^2 F. For normal incidence of the sun's rays on ordinary window glass, $b = 0.04$ and $t = 0.88$. Then, the sol-air temperature for the glass in this case with the sun's rays at normal incidence is:

$$t_e = 90 + 300\left(\frac{0.04}{4} + \frac{0.88}{1.13}\right)$$

$$= 327 \text{ F}$$

The rate of heat transfer to the inside of the enclosure with $t_i = 80$ F, is, for this case,

$$\frac{q}{A} = 1.13\,(327 - 80) = 279 \text{ Btu/hr ft}^2$$

For engineering purposes, since the second term on the right side of Equation 9 is small in comparison with the first, the contribution to the cooling load due to heat transfer through glass is, approximately,

$$\frac{q}{A} = tI \quad \text{(12)}$$

In the foregoing example, the use of this approximation would give a result of 264 Btu/hr ft^2 instead of the correct result of 279 Btu/hr ft^2.

(See p. 106 for Discussion.)

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No. 1268

SUMMER WEATHER DATA AND SOL-AIR TEMPERATURE—STUDY OF DATA FOR LINCOLN, NEBRASKA

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SUMMER WEATHER data taken by the United States Weather Bureau at its station in Lincoln, Nebr., during a ten-year period from 1932 through 1941, are used in this paper to determine design sol-air [1] temperatures for that city. This study was made at the suggestion of the A.S.H.V.E. Research Technical Advisory Committee on Cooling Load in Summer Air Conditioning.

The station at Lincoln is located at a north latitude of 40°50′, a west longitude of 96°45′ and an altitude of 1225 ft. It is believed that the solar radiation results obtained at this station are very representative of the Great Plains area. Observed data were (1) hourly readings of the dry-bulb temperature of the outdoor air in the shade, (2) hourly readings of the total solar and sky radiation incident upon a horizontal surface and (3) hourly wind movement. These data were obtained for each hour of the day during each day of the four months of June, July, August and September for the ten-year period from 1932 through 1941. For example, for the hour ending at 9 a.m. in the month of July, there were 310 readings of each of the three variables—air temperature, total solar and sky radiation and hourly wind movement.

The method of analysis of the data was the same as that followed in the earlier paper.[2] The sol-air temperature at each hour, for materials which do not directly transmit solar radiation, may be found from the equation:

$$t_e = t_a + \frac{bI}{4} \quad \dots \dots \dots \quad (1)$$

where

t_e = sol-air temperature, Fahrenheit;
t_a = temperature of the outdoor air in the shade, Fahrenheit;
b = absorptivity of the surface for solar radiation;
I = intensity of total solar and sky radiation incident upon the surface, Btu/hr ft².

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[1] Summer Weather Data and Sol-Air Temperature—Study of Data for New York City, by C. O. Mackey and E. B. Watson. (See p. 75 this volume.)
[2] Loc. Cit. Note 1.

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Strictly, a value of the outdoor air film coefficient that varied with wind velocity should be used in Equation 1 but, for simplicity, a constant value of 4 Btu/hr ft^{2}F was used. This question is discussed in Appendix A.

Temperatures of the outdoor air in the shade and sol-air temperatures for the horizontal surface are precise, since they depend upon observed data only. As explained in the earlier paper,[3] the calculation of sol-air temperatures for

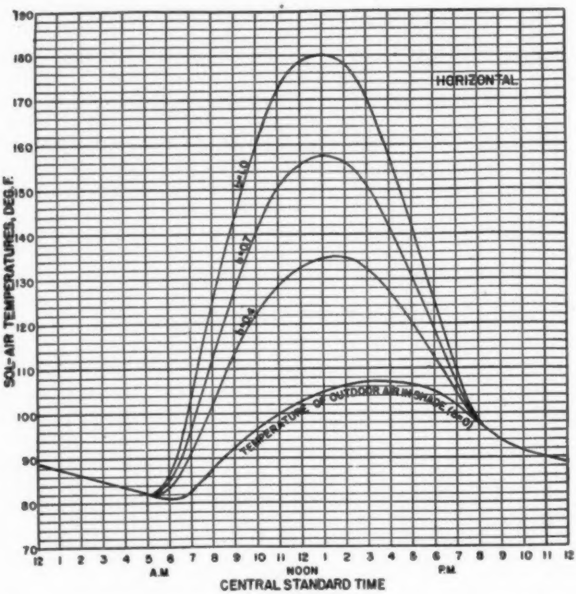

FIG. 1. DESIGN SOL-AIR TEMPERATURES FOR HORIZONTAL SURFACES IN LINCOLN, NEBRASKA

vertical surfaces of various orientations involves certain arbitrary assumptions, because the total solar and sky radiation incident upon these surfaces is not actually measured at the stations of the Weather Bureau. One assumption involves the ratio of direct to diffuse radiation received upon the horizontal surface as a function of solar altitude. The same graph of this function was used in this paper as in analyzing the New York data. Another assumption concerns the diffuse sky radiation incident upon vertical surfaces of different orientations at different times of the day; as in the earlier paper, it was assumed that the diffuse sky radiation incident on a vertical surface of any orientation is one-half of that incident upon a horizontal surface at the same time of day. These two assumptions are not exactly true, and the sol-air temperatures for vertical surfaces are not precisely correct. Their effect upon the

[3] Loc. Cit. Note. 1.

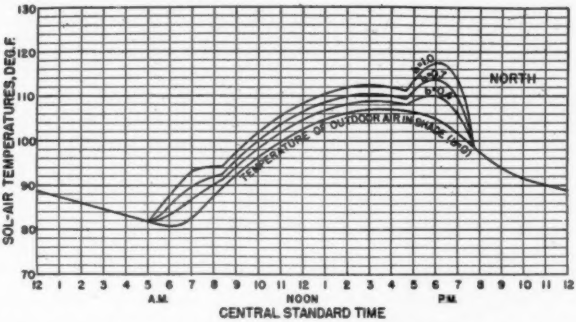

Fig. 2. Design Sol-Air Temperatures for Vertical Surface Facing North in Lincoln, Nebraska

Table 1—Outdoor Air Temperature Data—Lincoln, Nebraska

(Maximum hourly outdoor air temperature in the shade, and hourly outdoor air temperature equalled or exceeded on no more than 5 per cent of the days during a given month in the period from 1932 through 1941)

Time of Day (C.S.T.)	Temperature of the Outdoor Air in the Shade, Degree F							
	Maximum				Equalled or Exceeded Only 5 Per Cent of the Total Hours, 1932–1941			
	June	July	August	Sept.	June	July	August	Sept.
1 AM	87	98	92	87	81	88	84	81
2	87	96	90	85	81	86	83	80
3	88	94	89	84	79	84	82	80
4	87	92	89	84	79	84	81	78
5	85	91	88	82	77	82	80	77
6	84	91	86	82	77	81	80	76
7	84	93	86	85	79	82	80	75
8	88	95	89	87	82	88	84	80
9	93	99	94	90	87	93	88	83
10	95	103	98	92	90	96	92	87
11	99	106	103	95	93	100	96	90
12	101	109	105	98	96	102	99	92
1 PM	105	112	107	102	98	104	101	95
2	107	113	108	103	99	106	103	97
3	108	115	110	105	100	107	103	97
4	107	114	110	105	100	107	103	97
5	106	113	109	105	99	106	103	95
6	105	112	108	102	97	105	101	92
7	103	108	106	99	96	102	98	90
8	99	105	102	96	92	98	94	89
9	95	103	98	93	88	94	91	85
10	93	100	96	92	85	92	89	84
11	92	99	95	91	84	90	87	83
12	90	99	93	89	83	89	85	82
24-hr average of above	95.3	102.5	98.0	93.0	88.4	94.4	91.1	86.0

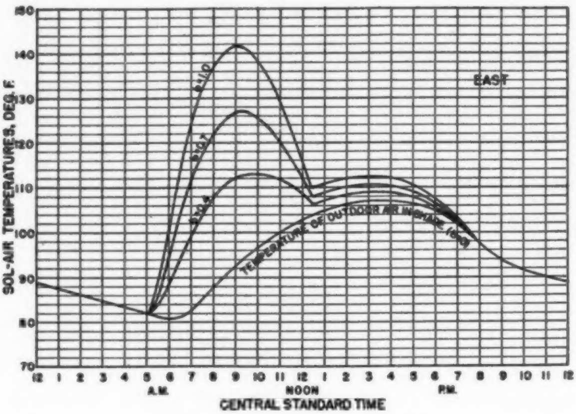

FIG. 3. DESIGN SOL-AIR TEMPERATURES FOR VERTICAL SURFACES FACING EAST IN LINCOLN, NEBRASKA

TABLE 2—INTENSITY OF TOTAL SOLAR AND SKY RADIATION ON A HORIZONTAL SURFACE—LINCOLN, NEBRASKA

(Maximum hourly intensity, and hourly intensity equalled or exceeded on no more than 5 per cent of the days during a given month in the period from 1932 through 1941)

TIME OF DAY (C.S.T.)	INTENSITY OF TOTAL SOLAR AND SKY RADIATION ON HORIZONTAL PLANE, BTU/HR FT²							
	MAXIMUM				EQUALLED OR EXCEEDED ONLY 5 PER CENT OF THE TOTAL HOURS, 1932–1941			
	June	July	August	Sept.	June	July	August	Sept.
5 AM	7	4	1	0	3	2	0	0
6	61	37	25	4	37	31	16	2
7	110	102	106	44	99	92	70	37
8	179	170	161	108	166	158	133	101
9	254	229	211	175	226	219	195	164
10	297	294	272	230	280	270	251	218
11	330	323	311	271	315	305	288	259
12	341	352	336	293	328	317	303	280
1 PM	348	356	332	295	333	320	306	283
2	328	337	322	281	317	308	286	263
3	285	293	270	242	275	270	247	220
4	247	230	217	188	225	218	196	167
5	176	169	157	117	166	163	137	106
6	110	110	88	47	101	96	71	38
7	67	50	31	10	41	37	18	4
8	16	11	4	0	7	5	2	0

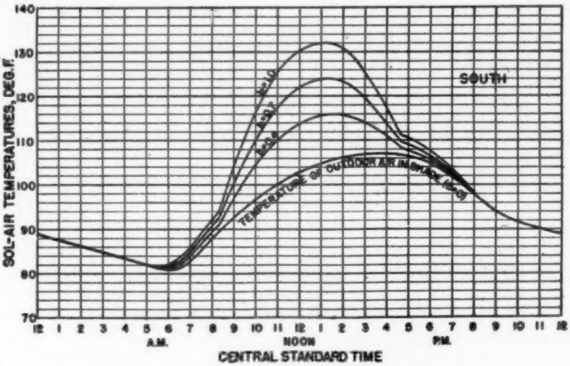

FIG. 4. DESIGN SOL-AIR TEMPERATURES FOR VERTICAL SURFACES FACING SOUTH IN LINCOLN, NEBRASKA

TABLE 3—MAXIMUM SOL-AIR TEMPERATURES FOR HORIZONTAL SURFACES IN LINCOLN, NEBRASKA

(*Maximum values for a ten-year period, 1932–1941*)

Time of Day (C.S.T.) (Solar Absorptivity)	Sol-Air Temperature, Degree F											
	June			July			August			Sept.		
	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)
1 AM....	87	87	87	98	98	98	92	92	92	87	87	87
2	87	87	87	96	96	96	90	90	90	85	85	85
3	88	88	88	94	94	94	89	89	89	84	84	84
4	87	87	87	92	92	92	89	89	89	84	84	84
5	86	85	85	91	91	91	88	88	88	82	82	82
6	92	89	87	96	94	93	88	88	87	83	82	82
7	106	99	93	106	102	98	103	98	93	95	92	89
8	126	115	103	132	121	110	125	114	103	114	106	98
9	146	130	114	154	137	121	141	127	113	129	117	106
10	165	144	123	172	151	130	157	139	122	147	131	114
11	177	153	130	182	159	136	171	150	130	157	139	120
12	180	156	132	185	162	139	176	154	133	165	145	125
1 PM....	182	159	136	186	164	142	179	158	136	168	148	128
2	180	158	136	182	162	141	178	157	136	165	147	128
3	173	153	134	175	157	139	167	150	133	156	140	125
4	162	145	129	167	151	135	156	142	128	140	129	119
5	146	134	122	148	138	127	141	132	122	125	119	113
6	128	121	114	132	126	120	124	119	114	109	107	105
7	110	108	106	115	113	111	109	108	107	100	99	99
8	100	99	99	105	105	105	102	102	102	96	96	96
9	95	95	95	103	103	103	98	98	98	93	93	93
10	93	93	93	100	100	100	96	96	96	92	92	92
11	92	92	92	99	99	99	95	95	95	91	91	91
12	90	90	90	99	99	99	93	93	93	89	89	89
24-hour average of above.	124.0	115.4	106.8	129.5	121.4	113.3	122.8	115.4	107.9	114.0	107.7	101.4

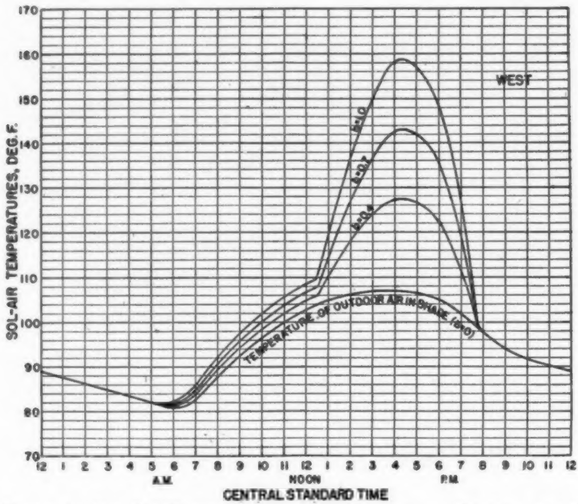

FIG. 5. DESIGN SOL-AIR TEMPERATURES FOR VERTICAL SURFACES FACING WEST IN LINCOLN, NEBRASKA

TABLE 4—SOL-AIR TEMPERATURES FOR HORIZONTAL SURFACES IN LINCOLN, NEBRASKA

(*Equalled or exceeded at any hour only 5 per cent of the total hours in a ten-year period, 1932–1941*)

TIME OF DAY (C.S.T.) (SOLAR ABSORPTIVITY)	DESIGN SOL-AIR TEMPERATURE, DEGREE F											
	June			July			August			Sept.		
	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)
1 AM....	81	81	81	88	88	88	84	84	84	81	81	81
2	81	81	81	86	86	86	83	83	83	80	80	80
3	79	79	79	84	84	84	82	82	82	80	80	80
4	79	79	79	84	84	84	81	81	81	78	78	78
5	77	77	77	82	82	82	80	80	80	77	77	77
6	82	81	79	87	85	83	82	81	81	76	76	76
7	98	93	87	103	97	91	95	90	86	83	81	78
8	119	108	97	124	113	102	114	105	96	101	95	89
9	138	123	107	143	128	113	134	120	106	121	110	98
10	155	136	116	160	141	122	151	133	116	137	122	107
11	166	144	122	172	151	129	164	144	123	150	132	114
12	173	150	127	178	155	133	171	149	128	157	137	118
1 PM....	176	153	129	180	157	134	173	151	130	161	141	121
2	173	151	129	178	156	135	170	150	130	157	139	121
3	165	145	126	170	151	132	161	144	126	148	132	117
4	151	136	120	158	143	127	149	135	121	133	122	112
5	135	124	114	142	131	120	133	124	115	117	111	104
6	119	113	106	126	120	113	116	111	107	99	97	95
7	104	101	99	109	107	105	101	100	99	90	90	90
8	92	92	92	99	98	98	94	94	94	89	89	89
9	88	88	88	94	94	94	91	91	91	85	85	85
10	85	85	85	92	92	92	89	89	89	84	84	84
11	84	84	84	90	90	90	87	87	87	83	83	83
12	83	83	83	89	89	89	85	85	85	82	82	82
24-hr. avg. of above..	116.0	107.7	99.5	121.6	113.4	105.3	115.3	108.1	100.8	106.3	100.2	94.1

sol-air temperature, however, is not great and the results are believed to be sufficiently accurate for the purpose of estimating the influence of heat transfer through building materials upon cooling load. This question is discussed at some length in the earlier paper.

TABLE 5—SOL-AIR TEMPERATURES FOR VERTICAL SURFACES IN LINCOLN, NEBRASKA

(Equalled or exceeded at any hour in July only 5 per cent of the total hours in a ten-year period, 1932–1941)

TIME OF DAY (C.S.T.) (SOLAR ABSORPTIVITY)	DESIGN SOL-AIR TEMPERATURE, DEGREE F											
	Surface Facing											
	North			East			South			West		
	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)	(1.0)	(0.7)	(0.4)
1 AM....	88	88	88	88	88	88	88	88	88	88	88	88
2	86	86	86	86	86	86	86	86	86	86	86	86
3	84	84	84	84	84	84	84	84	84	84	84	84
4	84	84	84	84	84	84	84	84	84	84	84	84
5	82	82	82	82	82	82	82	82	82	82	82	82
6	88	87	84	100	95	89	82	82	82	82	82	82
7	93	90	87	125	112	99	85	84	83	85	84	83
8	94	92	91	137	122	108	92	91	90	92	91	90
9	98	97	95	142	127	112	104	101	97	98	97	95
10	102	100	98	138	126	113	115	110	104	102	100	98
11	106	104	102	129	120	112	125	118	110	106	104	102
12	108	106	104	115	111	107	130	122	113	108	106	104
1 PM....	110	108	106	110	108	106	132	124	115	119	115	110
2	112	110	108	112	110	108	131	123	116	137	128	118
3	113	111	109	113	111	109	126	120	114	150	137	124
4	112	111	109	112	111	109	117	114	111	158	143	127
5	113	111	109	110	109	108	110	109	108	157	141	126
6	117	114	110	108	107	106	108	107	106	149	136	123
7	113	110	107	104	103	103	104	103	103	128	120	112
8	98	98	98	98	98	98	98	98	98	98	98	98
9	94	94	94	94	94	94	94	94	94	94	94	94
10	92	92	92	92	92	92	92	92	92	92	92	92
11	90	90	90	90	90	90	90	90	90	90	90	90
12	89	89	89	89	89	89	89	89	89	89	89	89
24-hour average of above..	98.6	97.4	96.1	105.9	102.5	99.0	102.1	99.8	97.5	106.6	102.9	99.3

Tables 1-5 and Figs. 1-5 are largely self-explanatory. Maximum temperatures and the temperatures equalled or exceeded only 5 per cent of the total hours in the ten-year period are tabulated for each hour of the day during each of the four months. For example, in Table 1, the maximum temperature of the outdoor air observed at 9 a.m. during the month of June (1932 through 1941) was 93 F, while in only 15 out of 300 readings at this hour did this temperature equal or exceed 87 F.

Although the average incident solar radiation is higher during June than July, the higher July air temperatures cause the sol-air temperatures to be

higher in July than in any other month. This is generally true for the latitudes of Lincoln and New York. Consequently, the *design* value of the sol-air temperature at any given hour has been taken as that temperature equalled or exceeded only 16 times in 310 observations for the month of July in the ten-year period from 1932 through 1941. The same procedure was followed in obtaining design sol-air temperatures from the New York data.

TABLE 6—DIRECT SOLAR RADIATION RATIOS AT LINCOLN, NEBRASKA (CALCULATING TABLE)
(*Solar declination of 21° 36′, mid-July*)

TIME (C.S.T.)	RATIO: DIRECT SOLAR RADIATION ON VERTICAL SURFACE TO DIRECT SOLAR RADIATION ON HORIZONTAL SURFACE			
	Vertical Surface Facing			
	North	East	South	West
6 A.M.	2.215	5.844	...	...
7	0.5600	2.693	...	...
8	0.07435	1.648	...	...
9	...	1.080	0.1447	...
10	...	0.6916	0.2600	...
11	...	0.3853	0.3200	...
12 noon	...	0.1163	0.3462	...
1 P.M.	...	...	0.3449	0.1424
2	...	...	0.3156	0.4138
3	...	...	0.2508	0.7257
4	...	...	0.1286	1.1260
5	0.1059	...	...	1.722
6	0.6405	...	...	2.855
7	2.617	...	...	6.584

TABLE 7—SOLAR ALTITUDE AT LINCOLN, NEBRASKA (CALCULATING TABLE)

TIME (C.S.T.)	SOLAR ALTITUDE NORTH LATITUDE OF 40° 50′ WEST LONGITUDE OF 96° 45′ SOLAR DECLINATION OF 21° 36′
5:07 A.M.	0° (Sunrise)
6:00	9° 5′
7:00	19° 59′
8:00	31° 13′
9:00	42° 33′
10:00	53° 33′
11:00	63° 24′
12:00 noon	69° 56′
1:00 P.M.	69° 32′
2:00	62° 30′
3:00	52° 29′
4:00	41° 25′
5:00	30° 5′
6:00	18° 52′
7:00	8° 2′
7:47	0° (Sunset)

ACKNOWLEDGMENTS

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APPENDIX A

WIND MOVEMENT AND SOL-AIR TEMPERATURE

In calculating the sol-air temperatures of this report, a constant value of the outdoor air film coefficient of heat transfer of 4 Btu/hr ft^2 F has been used at each hour regardless of the actual wind velocity. According to the results of Rowley, Algren, and Blackshaw,[4] this value is approximately correct for a *parallel* movement of air past a surface (mean temperature of surface and air of 80 F) when the wind velocity is 5 mph with a brick or concrete surface (rough), and 10 mph for a glass or painted wood surface (smooth).

The hourly wind movement observed during the month of July at Lincoln, Nebraska, over a five-year period is summarized in Table A-1.

[4] A.S.H.V.E. Research Report No. 869—Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw. (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 429.)

The *maximum* hourly wind movement observed in the same period was 19 miles for the hour ending at 9 a.m., 22 miles for the hour ending at 12 noon and 20 miles for the hour ending at 3 p.m. The minimum hourly wind movement observed at Lincoln, Nebraska, in July during the same period was 3 miles for the hours ending at 9 a.m. and at 12 noon and 4 miles for the hour ending at 3 p.m.

A slight tendency was observed for the higher values of the total solar and sky radiation to occur simultaneously with the higher wind velocities. This effect, however, is not invariable. For example, the solar and sky radiation incident upon the

TABLE A-1—HOURLY WIND MOVEMENT OBSERVED DURING MONTH OF JULY AT LINCOLN, NEBRASKA, OVER FIVE-YEAR PERIOD

YEAR	AVERAGE HOURLY WIND MOVEMENT IN MILES DURING JULY FOR HOUR ENDING		
	9 A.M.	Noon	3 P.M.
1932 (31 readings)	10.5	10.2	11.1
1933 (31 readings)	8.5	9.1	9.6
1934 (31 readings)	10.3	11.6	11.8
1935 (31 readings)	9.0	10.1	11.2
1936 (31 readings)	10.3	11.8	11.9
1932–1936 (155 readings)	9.7	10.5	11.1

horizontal was 352 Btu/hr ft^2 on July 21, 1933, while the wind velocity was 7 mph for the hour ending at 12 noon; on July 5, 1933, the solar and sky radiation incident upon the horizontal for the same hour was 341 Btu/hr ft^2 with a wind velocity of 16 mph.

Although the value of the outdoor air film coefficient of heat transfer used in calculating the sol-air temperature from weather data should vary with the instantaneous wind velocity for perfection, this procedure is complicated. In the light of the above facts, the procedure of using a constant value of 4 for this coefficient will yield reasonably reliable values that will, in general, be on the safe side when used for estimates of cooling loads.

APPENDIX B

COMPARISON OF SOL-AIR TEMPERATURES IN NEW YORK, N. Y., AND LINCOLN, NEBRASKA

The sol-air temperatures that were equalled or exceeded only five per cent of the total hours during July in the ten-year period from 1932 through 1941 are used as *design* values; *average* daily design values may be compared for New York and Lincoln (see Table B-1).

The average daily design sol-air temperatures in Lincoln, Nebraska, are from 9.6 F to 15.2 F higher than the corresponding temperatures in New York, N. Y.; the actual difference depends upon the orientation of the surface and its solar absorptivity.

Since the latitudes of these two cities are nearly the same, the sol-air temperature *vs.* time of day curves have about the same shape. One way of comparing these shapes for a given orientation and absorptivity is to compare the ratios of the difference between the sol-air temperature at a given time and the average daily sol-air temperature, to the daily range in the sol-air temperature. Such a comparison appears in Table B-2 for a horizontal surface with a solar absorptivity of 0.7.

On a coordinate system with the ratio of the difference between the sol-air temperature at given time and the daily sol-air temperature to the daily range in sol-air temperature $\left(\frac{t_e - t_m}{\Delta t_e}\right)$ plotted against time of day, one curve will closely fit the

TABLE B-1—COMPARISON OF AVERAGE DAILY DESIGN VALUES FOR JULY IN NEW YORK AND LINCOLN

DESIGN SOL-AIR TEMPERATURES, F	NEW YORK, N. Y.	LINCOLN, NEBR.
Average daily outdoor air in shade (b = 0)	84.8	94.4
Average daily temp. for horizontal surface (b = 0.4)	93.2	105.3
Average daily temp. for horizontal surface (b = 0.7)	99.8	113.4
Average daily temp. for horizontal surface (b = 1.0)	106.4	121.6
Average daily temp. for vertical surface facing N (b = 0.4)	86.2	96.1
Average daily temp. for vertical surface facing N (b = 0.7)	87.3	97.4
Average daily temp. for vertical surface facing N (b = 1.0)	88.5	98.6
Average daily temp. for vertical surface facing E (b = 0.4)	87.8	99.0
Average daily temp. for vertical surface facing E (b = 0.7)	89.9	102.5
Average daily temp. for vertical surface facing E (b = 1.0)	92.3	105.9
Average daily temp. for vertical surface facing S (b = 0.4)	87.1	97.5
Average daily temp. for vertical surface facing S (b = 0.7)	89.0	99.8
Average daily temp. for vertical surface facing S (b = 1.0)	90.7	102.1
Average daily temp. for vertical surface facing W (b = 0.4)	89.5	99.3
Average daily temp. for vertical surface facing W (b = 0.7)	93.0	102.9
Average daily temp. for vertical surface facing W (b = 1.0)	96.5	106.6

TABLE B-2—COMPARISON OF RATIO: DIFFERENCE BETWEEN THE SOL-AIR TEMPERATURE AT A GIVEN TIME AND THE AVERAGE DAILY SOL-AIR TEMPERATURE, TO THE DAILY RANGE IN THE SOL-AIR TEMPERATURE

TIME OF DAY	DESIGN SOL-AIR TEMPERATURE, t_e		RATIO: $\frac{t_e - t_m}{\text{RANGE OF } t_e}$	
	New York, N. Y.	Lincoln, Nebr.	New York, N. Y.	Lincoln, Nebr.
1 AM	78	88	−0.35	−0.34
2	77	86	−0.38	−0.37
3	77	84	−0.38	−0.39
4	76	84	−0.40	−0.39
5	76	82	−0.40	−0.42
6	79	85	−0.35	−0.38
7	92	97	−0.13	−0.22
8	102	113	+0.04	0.00
9	114	128	0.24	+0.20
10	122	141	0.37	0.37
11	130	151	0.50	0.50
12 noon	135	155	0.59	0.56
1 PM	136	157	0.60	0.58
2	124	156	0.57	0.57
3	129	151	0.49	0.50
4	123	143	0.38	0.40
5	112	131	0.20	0.23
6	101	120	0.02	0.09
7	91	107	−0.15	−0.08
8	85	98	−0.25	−0.20
9	83	94	−0.28	−0.26
10	82	92	−0.30	−0.29
11	81	90	−0.31	−0.31
12	79	89	−0.35	−0.33
24-hour average, t_m	99.8	113.4		
Daily range, Δt_e	60	75		

results observed at Lincoln and at New York for a given orientation and solar absorptivity of the surface. Latitude, however, would be expected to have an effect upon the shapes of these curves; in other words, corresponding curves for New Orleans, for example, would probably not have the same shape.

The effect of the greater sol-air temperature in Lincoln, Nebraska, upon the maximum rate of heat flow through a typical wall is next shown by an example. Assume a homogeneous, vertical, west wall made of 8 in. of brick with a solar absorptivity of the outside surface of 0.7. By the approximate method of solution which uses a decrement factor and time lag that are the arithmetic means of the values for the fundamental and second harmonic,[5] the average decrement factor for this wall is:

$$\lambda = \frac{0.15 + 0.08}{2} = 0.115,$$

and the average time lag is:

$$\phi = \frac{\frac{83}{15} + \frac{131}{30}}{2} = 5.0 \text{ hour}$$

For this west wall, the maximum design sol-air temperature in *New York, N. Y.*, is 129 F at 4:00 p.m. Consequently, for a constant temperature of the indoor air of 80 F, the maximum temperature of the inside surface of this wall would be 87.4 F at 9:00 p.m.; the corresponding maximum rate of heat transfer at the indoor surface is *12.2 Btu/hr ft²*. In *Lincoln, Nebraska,* the maximum design sol-air temperature for this wall is 143 F at 4:24 p.m.; for a constant temperature of the indoor air of 80 F, the maximum temperature of the inside surface of this wall would be 90.3 F at 9:24 p.m., and the maximum rate of heat transfer is *17.0 Btu/hr ft²*. In the case of this one wall, the greater sol-air temperatures at Lincoln, Nebraska, cause the maximum rate of heat transfer to be 40 per cent higher than in New York, N. Y. The principal reason for this is the greater daily average design sol-air temperature for this surface in Lincoln (102.9 F *vs.* 93.0) rather than the greater daily range in design sol-air temperature (61 F *vs.* 54 F). If the temperature of the indoor air were held constant at 80 F for 24 hours, the quantity of heat transmitted daily through one square foot of this same wall would be 224 Btu in Lincoln as against 127 Btu in New York on the design day. The *maximum* rate of heat transfer affects the required capacity of the cooling unit, but the *daily* rate of heat transfer affects the daily energy consumption of that unit.

The sol-air temperature study would seem to indicate that the load on the cooling equipment in Lincoln, Nebraska, and the energy consumption for the design conditions, insofar as they are affected by solar radiation and air temperature alone in maintaining a constant temperature of the indoor air of 80 F are substantially the same as if a constant indoor air temperature of 70 F were to be maintained in New York, N. Y.

APPENDIX C

OTHER SOL-AIR TEMPERATURES IN U.S.A.

Design sol-air temperatures have now been found by the same method for Lincoln, Nebraska, and for New York, N. Y. In *Monthly Weather Review,* April, 1941, Dr. I. F. Hand gives interesting data on the measurement of total solar and sky radiation. Some of the pertinent results given in this reference or derived from it are presented.

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[5] A.S.H.V.E. Research Report No. 1255—Periodic Heat Flow-Homogeneous Walls or Roofs, by C. O. Mackey and L. T. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 293.)

The departure of the average total solar and sky radiation received daily at each station in a given band of latitudes from the composite for that band follows:

35°–45° N. Latitude—(Composite of 8 stations for week beginning July 2)

	PER CENT
Fresno, Cal.	+25.2
Washington, D. C.	− 5.1
New York, N. Y.	− 5.4
Lincoln, Nebr.	+ 3.9
Chicago, Ill.	−19.0
Blue Hill, Mass.	+ 3.3
Twin Falls, Idaho	− 3.4
Madison, Wis.	+ 0.1

Except for Fresno, where there is an exceedingly high percentage of sunshine (San Joaquin Valley), and Chicago, where very smoky conditions prevail near the station, the departure of each station from the average is not very great.

There may be a correlation between design sol-air temperatures and these departures, but data have not been secured from enough cities to check this. It has been pointed out that the difference between average daily design sol-air temperatures in Lincoln, Nebr., and New York, N. Y., is about 10 F for all orientations and solar absorptivities. Based on the average radiation incident upon the horizontal for the week of July 2, 9.3 percentage points separate Lincoln and New York in the departure table; based on the average radiation received during the entire year, 20 percentage points separate the same two stations in the departure table. Whether the correlation, if any, is between design sol-air temperature and summer departure or annual departure may only be found from more data.

It is proposed to obtain data for the month of July from two more stations, Washington and Chicago, and for the months of June, July and August from one station in the lower band of latitudes, perhaps New Orleans, if the committee deems this desirable. It no longer seems necessary to analyze the data for four months as was done in Lincoln and New York and this will cut down on the time required for the study.

TABLE C-1—PYRHELIOMETRIC STATIONS IN U. S. A.

(*Arranged in order of latitudes*)

STATION	UNDER DIRECTION OF	N. LAT.		W. LONG.		ALTITUDE, FT
		Deg	Min	Deg	Min	
San Juan, P. R.	U.S.W.B.	18	28	06	06	85
Miami, Fla.	Dr. O. J. Sieplein	25	41	80	12	50
New Orleans, La.	Tulane University	29	56	90	07	100
La Jolla, Cal.	Scripps Inst. of Oceanography	32	50	117	15	85
Riverside, Cal.	University of California	33	58	117	28	1051
Fresno, Cal.	U.S.W.B.	26	43	119	49	330
Washington, D. C.	U.S.W.B.	38	56	77	05	397
New York, N. Y.	U.S.W.B.	40	46	73	58	180
Lincoln, Nebr.	U.S.W.B.	40	50	96	45	1225
Chicago, Ill.	U.S.W.B.	41	47	87	25	688
Blue Hill, Mass.	Harvard	42	13	71	07	640
Twin Falls, Idaho	U. S. Bureau of Entomology	42	29	114	25	4300
Madison, Wis.	U.S.W.B.	43	05	89	23	974
Friday Harbor, Wash.	University of Washington	48	32	123	01	15
Fairbanks, Alaska	U.S.W.B.	64	52	147	39	500

TABLE C-2—MEAN HOURLY TOTALS OF SOLAR AND SKY RADIATION FOR WEEK BEGINNING JULY 2

(Received on a horizontal surface for stations at north latitudes between 35 deg and 45 deg—apparent solar time)

STATION	BTU/FT² FOR HOUR ENDING														TOTAL 6 A.M.-7 P.M.
	A.M.							P.M.							
	6	7	8	9	10	11	12	1	2	3	4	5	6	7	
Fresno (1936–39)..	26	89	159	218	266	299	318	318	292	255	210	151	81	26	2708
Washington (1927–39)..	21	66	115	159	197	224	242	240	231	204	153	112	65	24	2053
New York (1936–39)..	30	74	111	162	207	229	229	229	222	192	159	114	66	22	2046
Lincoln (1925–39)..	22	66	122	173	214	247	266	266	255	218	173	126	74	26	2248
Chicago (1923–39)..	20	54	94	134	167	188	198	204	198	167	144	103	59	22	1752
Blue Hill (1936–39)..	30	74	125	170	218	247	266	269	251	206	173	118	66	22	2235
Twin Falls (1936–39)..	33	74	118	159	195	229	240	244	214	188	159	122	81	33	2089
Madison (1936–39)..	22	66	118	155	199	244	258	266	247	214	173	118	70	26	2176
Average for 8 stations..															2163

TABLE C-3—MEAN HOURLY TOTALS OF SOLAR AND SKY RADIATION

(Received on a horizontal surface for week beginning as indicated for stations at north latitudes between 25 deg and 35 deg—apparent solar time)

STATION	BTU/FT² FOR HOUR ENDING														TOTAL 6 A.M.-7 P.M.
	A.M.							P.M.							
	6	7	8	9	10	11	12	1	2	3	4	5	6	7	
Miami (1936–38) (June 4)	4	30	78	136	184	206	232	236	225	192	162	111	63	26	1885
New Orleans (1936–39) (June 11)..	7	44	103	159	206	240	262	258	218	177	137	103	52	22	1988
La Jolla (1936–39) (June 4)...	15	44	96	151	214	269	295	291	273	229	170	111	70	11	2139
Riverside (1936–39) (June 11)..	11	55	107	166	214	258	280	276	258	225	181	122	59	15	2227
Average for 4 Stations..															2085

25°–35° N. Latitude—(Composite of 4 stations for week beginning June 4 or June 11)

	PER CENT
Miami, Fla.	−9.6
New Orleans, La.	−4.7
La Jolla, Cal.	+7.4
Riverside, Cal.	+6.8

The departure of the average daily solar and sky radiation received during the entire year from the composite for ten stations follows:

	PER CENT
Miami, Fla.	+ 9
New Orleans, La.	+ 2
[a]La Jolla, Cal.	+16
[a]Riverside, Cal.	+14
[a]Fresno, Cal.	+29
[a]Washington, D. C.	− 5
[a]New York, N. Y.	−16
[a]Lincoln, Nebr.	+ 4
[a]Chicago, Ill.	−24
Blue Hill, Mass.	− 5
[a]Twin Falls, Idaho	+ 4
[a]Madison, Wis.	− 5

[a] These stations and Friday Harbor were included in the composite.

DISCUSSION

JOHN EVERETTS, JR., San Francisco, Calif. (WRITTEN): The authors of this paper are to be complimented upon their study and presentation of a subject which has heretofore been quite nebulous and irrational in spite of its importance to the air conditioning engineer.

The effect of solar radiation on an air conditioned space has been guessed at for years. In most cases, the guess has been conservative or the solar radiation load has been small compared to the total load and, consequently, the results have not been disastrous.

The only point to which I wish to take exception is the statement attributed to Dr. I. F. Hand, U. S. Weather Bureau, that he believes solar radiation values for New York City are representative of average large city conditions. This is not true.

From the study of weather data made by Albright [6] and Everetts [7] from 1931-1937, it was found that dust and dirt contamination in the atmosphere had little effect upon the solar radiation as compared to the effect of haze and light fog.

A recheck of 12 cities where solar intensities are recorded shows the following comparison with New York City, taken at a value of 100 per cent:

East: Washington, D. C.—103 per cent; Boston, Mass.—97 per cent; Ithaca, N. Y.—100 per cent; Newport, R. I.—99 per cent; State College, Pa.—104 per cent.

Midwest: Nashville, Tenn.—113 per cent; Madison, Wis.—111 per cent; Lincoln, Nebr.—117 per cent.

West Coast (South): Fresno, Calif.—151 per cent; Davis, Calif.—156 per cent; Riverside, Calif.—134 per cent.

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[6] Analysis of Summer Weather Data in the United States, by J. C. Albright. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 397.)

[7] Application of Summer Weather Data in Design, by John Everetts, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 431.)

These percentages are based upon the mean solar intensity for the months of June, July, August and September and are indicative of a trend based upon factors other than latitude, physical size of the city and atmospheric contamination.

The material contained in this paper is of such importance that I would like to submit two recommendations for consideration by the Society:

1. That the data contained in the subject paper be studied by the Technical Advisory Committee on Cooling Load in Summer Air Conditioning to determine the practical value of these data as applied to design.
2. That, if these data are found useful in design, the authors be requested to continue their studies for those cities where solar radiation readings are taken and that the work be done in conjunction with the Technical Advisory Committee on Weather Design Data.

I would like to take this opportunity to thank Professor Mackey and Professor Watson for their valuable contribution to the Society.

G. V. PARMELEE,[8] Cleveland, Ohio. (WRITTEN): The author's concept of the sol-air temperature has been of considerable aid in the solution of periodic heat flow problems. A single variable now replaces the two variables, outdoor air temperature and solar intensity. The values are dependent upon the assumption of an outdoor air film coefficient which is ordinarily a function of wind velocity. This coefficient also includes the effect of radiation exchange between a surface and its surroundings.

It seems to me that the concept of sol-air temperature might be extended to include *all radiation* effects. The air film coefficient used would be strictly a function of velocity. The reasoning back of this statement is based upon the fact that the radiation and convective heat transfer are not at all times additive. Study of a paper [9] previously published yields some interesting data. For example, curves were drawn for a 2 in. concrete roof with a smooth asphalt finish. Between the hours of 9:00 p.m. and 6.00 a.m. the temperature difference between the air and the top surface of the roof ranged from zero to one degree, averaging about one-half degree. The wind velocity averaged about 4 mph during this period. The average temperature of the roof dropped steadily one degree or more per hour. Based upon the heat capacity of 6.92 Btu per (sq ft) (degree) as given in a paper [10] published in 1942 on the same subject, heat stored in the roof was being lost at the rate of 7 to 20 Btu per (sq ft) (hr).

During the early part of the period heat was being gained from the outdoor air at the rate of 3 or 4 Btu per (sq ft) (hr) and was lost from the under side of the roof at the rate of 4 to 9 Btu per (sq ft) (hr), while the roof lost stored at the rate of 14 to 25 Btu per (sq ft) (hr).

In the last part of this period heat was gained from the room at the rate of 2 or 3 Btu per hour and was lost to the outdoor air at about the same rate. In brief, heat was being lost by the roof at a much greater rate than could be accounted for by application of commonly accepted film coefficients and the measured heat flow rates, because of the large heat loss by radiation to the night sky.

Heat gains in the early morning hours are not of great interest, but performance of a roof or wall during the day is certainly affected by its previous history. Possibly a better agreement between test results and theoretical calculations could be obtained if consideration were given to nocturnal radiation effects. Although the clear night sky is sometimes considered as a black body at −40 F, some experimental work should be carried out to accurately measure the magnitude of this effect.

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[8] Research Fellow, Fenn College.

[9] Summer Cooling Load as Affected by Heat Gains Through Dry, Sprinkled and Water Covered Roofs, by F. C. Houghten, H. T. Olson and Carl Gutberlet. (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 231.)

[10] A.S.H.V.E. Research Report No. 1195—Heat Gain Through Walls and Roofs as Affected by Solar Radiation, by F. C. Houghten, E. C. Hach, S. I. Taimuty and Carl Gutberlet. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 91.)

VICTOR PASCHKIS,[11] New York, N. Y. (WRITTEN): The approximate method described by Mackey and Wright [12] is based on the use of the fundamental harmonic in Fourier's analysis. In simple words, in the approximate solution an equivalent sine wave curve for the sol-air temperature is used instead of the actual curve. As shown by the authors in their Appendices A and C this causes an appreciable error; the maximum rate of heat transfer from the inside surface was 25 per cent higher than the actual maximum rate and the time of the maximum was one hour later than the actual time.

These discrepancies were found for a sol-air temperature as per Fig. 3 of this early paper.

Now the sol-air temperature as per Figs. 1 to 3 of the present paper appears to be less similar to a sine wave than Fig. 3 of the earlier paper. Therefore, it might be expected that the errors due to the approximate method might be larger than those in the original article. It would be interesting if the author in his closure would show to what extent it is still permissible to use the approximate method.

W. H. CARRIER, Syracuse, N. Y.: I enter this discussion with rather mixed feelings. My first is an admiration for the ingenious device which the authors have called the *sol-air* temperature. My second reaction is that the *sol-air* temperature effect is of relatively little importance in relation to the total load. For example, the total load over the entire building due to sunlight, including the *sol-air* effect and windows, may be less than 15 per cent of the total load. Of this 15 per cent, less than 25 per cent can be traceable to *sol-air* effect on the walls where they are of at least equivalent to 8 in. masonry construction. In other words, probably less than 4 per cent of the entire load on the average masonry building is traceable to *sol-air* effect. Any such percentage as this is readily covered by normal factors of safety. In buildings with 12 in. masonry walls, the *sol-air* effect does not appear during the peak load but generally about 12 hours after the maximum *sol-air* temperature when no cooling is taking place. In the warmer climates, where there is not much cooling at night, there is some residual effect of heat remaining in the walls, but tests have shown, in locations such as in Detroit, that at the time of peak load the effect of *sol-air* temperature is practically zero.

Therefore, from the standpoint of sizing apparatus, in northern climates at least, the *sol-air* temperature effect can probably be neglected if the walls are of 8 in. masonry. The temperature lag effect, which the authors have also studied and reported in previous papers, will determine whether the *sol-air* effect need be taken into consideration in determining the size of equipment. In other words, due to lag this effect may or may not make an addition to the peak load.

In southern climates, such as in Texas, there is a positive wall load which, for most constructions, is additive to the peak load but not to the extent of anything like the maximum *sol-air* correction. It has been my practice, where there were 12 in. walls, to approximate the peak load by taking the *average* temperature for 24 hours as the outside temperature and adding to it the *average* sun effect as calculated for 24 hours. I made such a calculation, for example, for a building in Shanghai where the climate approximates that of southern Texas and I believe the total differential to be applied for the walls to determine the peak load was equivalent to about 8 deg between inside and outside temperatures, a rather surprisingly low figure.

However, when we consider the effect on the roof, the *sol-air* temperature becomes very much more important, unless there is an attic space or exceptional insulation such as 2 in. corkboard. On this particular surface, there is an accumulative effect for practically 12 hours of the day with the most direct angle of the sun's rays. Also in conjunction with this, there is a relatively thin structure and, therefore, a very small amount of time lag. In such structures, which do not make provision for dead air space or extra good insulation, the *sol-air* effect may become a very impor-

[11] Research Associate, Dept. of Mechanical Engineering, Columbia University.
[12] Loc. Cit. See Note 5.

tant element on the top floor and a considerable item in determining the maximum size of equipment to employ for the building.

The ingenious character of the *sol-air* temperature, I think, has not been sufficiently emphasized by the authors and others. The essential characteristic of this fictitious temperature is that it is not affected by the character of the walls whether they be sheet metal, 12 in. brick, or 2 in. cork. Furthermore, it supplies the correct answer regardless of the lag in heat transfer which may occur through the wall. In other words, the *sol-air* temperature offers an invariable basis for the calculation of heat transfer for all wall structures except glass.

This *sol-air* temperature is a unique thing and its invariability is not self-evident as one might superficially think. It has been mathematically proven by the authors to give the exact equivalent results. This proof becomes evident only by a study of their heat transfer equation. This is a very important discovery and the authors are to be complimented on inventing this fictitious temperature which permits an exact measure of heat transfer, regardless of other variables.

L. T. AVERY, Cleveland, Ohio: This paper will teach us more respect for sun load, and I think we should be more respectful. Most air-conditioning systems fail in performance when the sun shines in the afternoon on the west side. Whether you can use these figures or not for sol-air through glass, Example E, I would remind you that the calculations show that the normal west window would transmit 279 Btu per (hr) (sq ft).

C. M. ASHLEY, Syracuse, N. Y.: I believe that this represents a very substantial contribution to our knowledge and it deserves a lot of study on the part of everyone who is interested in the subject because it shows that the previous conceptions as to the nature of the sunlight radiation have been wrong in many respects.

For instance at just an hour before sundown; the solar radiation in New York is shown to be approximately 160 Btu per (hr) (sq ft). In Nebraska it is 124 and THE GUIDE shows it to be about 50. It seems to me that we ought to do a little work on THE GUIDE values.

It is interesting to note the difference in shape between the New York and the Nebraska curves. The values for New York are much smaller in the morning than for Nebraska and somewhat larger in the afternoon.

Professor Watson's point is very well taken; from an absolute point of view there is negative radiation in the night; but from a relative point of view I do not believe it has very much significance. It does deserve further study.

PROFESSOR MACKEY: It is possible that you did not understand the statement made by my colleague about the method of combining the air temperature and the solar radiation. He stated that the maximum air temperature has been found not to occur at the same time as the maximum solar radiation. That means not on the same day. We have air temperatures at a certain hour and solar radiation at a certain hour but we only combine them to obtain design values of the sol-air temperatures when they occurred on the same day at the same hour.

I wish to thank Commander Everetts for his discussion of the paper and for the correction regarding the quotation from Dr. Hand. We quoted that in good faith. It appears in that table in the *Monthly Weather Review* which gave characteristics at these stations where total solar and sky radiation is measured but I notice an apparent inconsistency. In the table, *35°-45° N. Latitude* (composite of 8 stations for week beginning July 2), in Appendix C of the Nebraska paper we point out that the departure of the average total solar and sky radiation received daily at each station in a given band of latitude from the composite of the band shows that New York, for example, in the band of the 35 to 45 deg north latitude for the week beginning July 2 is —5.4 per cent departure; that is, solar and sky radiation is below the composite of the band and Lincoln is +3.9.

Professor Parmelee's valuable discussion is appreciated. We know positively that he is right in his stand that there should be some correction to the sol-air tempera-

ture for the fact that there is a loss of heat from a building surface by radiation to a cold sky on clear nights. The effective temperature of space, as he says, is around —40 F and a building surface at night on clear nights radiates to space as if that space were at a temperature of 40 deg below zero on the Fahrenheit scale and that makes a considerable loss.

What that does to sol-air temperature is something that we have not included in our curves. It is the effect of a negative incident solar radiation and decreases the sol-air temperature below the air temperature for those hours of the night when this situation exists.

You might conclude that it is not important since it happened at night; but, since periodic repetition each day of this cycle is assumed, what happens at night also affects the heat gain when it reaches its peak the next day.

I wish to thank Dr. Carrier for his discussion. His criticism was well taken. We realize that the contribution to the cooling load from the heat transfer through building walls—it might not apply to roofs—is often a very small part of the total cooling load when the total cooling load consists of so many other factors, such as the load from lighting, the load from processes, the load from machinery, the load from people and the very important load from solar heat gain through window glass.

At the same time, if our method is correct, it will show just as well the insignificance of the solar heat gain through building walls and roofs as it will show its extreme significance, because with an accurate value you may then compare it with the other elements of the cooling load, which we wish could be calculated as accurately.

I wish to thank Mr. Avery for his emphasis on the importance of the sun load and Mr. Ashley for his discussion.

In closing the discussion I would like to emphasize again that I thank Dr. Carrier for his very kind and flattering remarks that *sol-air* temperature is a new concept. We believe it to be new. We found nowhere else the concept of combining air temperature and solar radiation in this way. Independently of its use in estimating cooling load it also has other applications. It may be used for discovering whether glass during the winter is a solar asset or a solar liability—that is, from the standpoint of heat gain through sun effect as against heat loss through conduction and it has many laboratory uses. In a room within a room, such as Dr. Winslow has at the Pierce Laboratory, and with no way of obtaining sunshine on a wall, it is possible to set up a cyclic air temperature and solar radiation without ever having to wait for the sun to shine with its design intensity on that wall.

No. 1269

ALTITUDE CHAMBER FOR STUDY OF HEATING AND AIR CONDITIONING PROBLEMS

By W. E. CROWELL,* BUFFALO, N. Y.

THE PURPOSE of this paper is not to give the answer to some problem but instead is to present problems which require answers. It is also presented to describe one device which is being widely used in solving these problems.

It may seem strange to start a discussion such as this by going back to the year 1862, but actually the first notable flight by man to high altitudes occurred in that year. Glaisher and Coxwell, two English balloonists, made an ascent in a balloon to approximately 29,000 ft and encountered severe physiological symptoms which later were to be the subject of much study by flight surgeons. As they ascended, Glaisher noticed his vision and hearing failing, next he became paralyzed in his legs and arms, and finally became unconscious. At the same time, his companion Coxwell encountered the same troubles, but with great presence of mind he pulled the dump valve with his teeth (his arms were paralyzed) and descended to a safe altitude.

A few years later, three other men, Tissandier, Crocé-Spinelli and Sivel, made a high altitude balloon flight which ended in partial disaster. All men lost consciousness at 26,000 ft but the balloon ascended to 28,820 ft and then descended on its own accord. Tissandier was the only one to recover.

Such flights attracted the attention of Paul Bert, a famous French physiologist, and he began a study of the effects on the human system of increased and decreased barometric pressures. As one of his laboratory devices, he used a steel tank large enough to accommodate a single person plus the necessary instruments. This, to the best of my knowledge, was the first altitude chamber. Today there are probably 40 or 50 large altitude chambers in this country alone and hundreds of smaller ones.

Up until the present war only a handful of men and machines had ever gone above 25,000 ft. Consequently, the information available concerning high altitude operation was scarce. With the coming of World War II, however, high altitude operations of bombers and fighters became a regular thing. But it was not a case of designing planes on the board, building them and sending them over Europe to operate in 500 or 1,000 plane raids. Our *Flying Fortresses* had been designed several years before the war and were supposed to be the best high altitude bombers in the world, yet when they first went into combat it was in very small numbers and the difficulties which arose were tremendous.

These problems have been studied in the past by two general methods of testing, flight tests and ground tests.

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Flight tests usually consist of installing the equipment to be studied in the airplane in the same manner as it will be installed when put in production, and then instrumenting it as much as is required to give accurate performance data and as well as the arrangement will permit.

Ground tests are of various types, some of which may be made with the apparatus installed in cold boxes, cold rooms or bell jars, size permitting. Instrumentation and control of conditions in general can be better than in flight

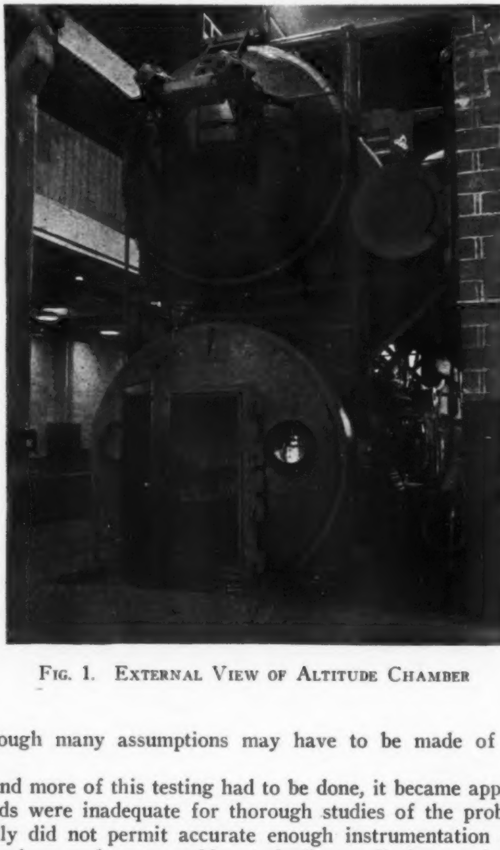

FIG. 1. EXTERNAL VIEW OF ALTITUDE CHAMBER

testing, although many assumptions may have to be made of some of the conditions.

As more and more of this testing had to be done, it became apparent that the above methods were inadequate for thorough studies of the problems. Flight tests generally did not permit accurate enough instrumentation or control of conditions, and ground tests could not duplicate all the required conditions. What was needed was a testing device which would exactly duplicate on the ground the high altitude conditions encountered in flight. Actually this would require a huge full scale wind tunnel in which air speeds, temperatures, pressures and humidities duplicating those at any desired altitude could be controlled. Obviously this would be an extremely costly piece of apparatus

beyond the price range which any aircraft manufacturer could afford. The majority of equipment on an airplane, however, is enclosed within the fuselage or wings and is not directly affected by outside airspeed. If airspeed can be eliminated, then the testing device can be built as an enclosure in which the other conditions are maintained as desired. This type of device is much less expensive than the do-all wind tunnel. Certain conditions to be met by such equipment are:

1. It must have complete instrumentation.
2. The desired conditions must be closely controlled.
3. Total operating cost must be as low as possible. It has been the experience of one aircraft company that experimental flight testing of large twin engine airplanes costs roughly $1000 overall for a 3 or 4 hour flight whereas the overall cost of operating a large altitude chamber is about $35 per hour.
4. The equipment must have all possible provisions for the safety of the operating personnel especially since it will often be necessary for some persons to be subjected to high altitude conditions.
5. Provisions should be made for making tests as quickly as possible. This will require that shop facilities be immediately available and that the conditions of operation be obtained quickly.
6. Provisions should be made so that wherever possible persons may observe the performance of the equipment being tested without subjecting themselves to high altitude conditions.
7. The apparatus must be suitable for teaching fliers how to use oxygen equipment and how to protect themselves at high altitudes. Simulated high altitude conditions in the chamber are needed to enable the fliers to apply the instructions received.

Several aircraft and aircraft equipment manufacturers have seen the need of such a device and have been attracted by the possibility of safer, less expensive, and more accurate high altitude testing. Since various companies had different ideas about the size of testing space, maximum conditions required, and limit of expenditure, it became apparent that each altitude chamber would be a tailor-made job. Smaller sized high altitude testing cabinets have become somewhat standardized, but no two of the larger sized altitude chambers are similar.

DESCRIPTION OF ALTITUDE CHAMBER

The altitude chamber at the Curtiss-Wright Research Laboratory in Buffalo is a good example of the larger sized chambers and a description of it is of interest.

Fig. 1 is an external view of the chamber showing the two shell arrangement. The upper shell, which is 10½ ft in inside diameter, contains all the evaporator coils and the cold air circulating fans driven by four variable speed induction motors, two of which are shown in the photograph. The lower shell or main test chamber is 10 ft in inside diameter and is connected to the upper shell by two large ducts. Each duct contains two sections for the cold air which is blown down from the evaporator coils and a center section through which the returning air can again pass into the upper shell. Dampers in these ducts actuated by a compressed air diaphragm motor accomplish the major part of the temperature control.

The large door shown in the end of the chamber is 4 ft x 7 ft and will allow passage of the complete fuselage of a pursuit type airplane. If the object

being installed in the chamber is too large to pass through this door, the whole end of the chamber may be removed.

Two of the observation ports of the main test chamber are shown at the end of the chamber. Five of these are 18 in. in diameter and one is 24 in. x 18 in. oval. Two ports of 10 in. diameter are in the small man lock at the opposite end of the chamber. Each port is built up of five plates of glass

FIG. 2. INTERIOR OF ALTITUDE CHAMBER

with a dehydrated air space between each plate. The main control panel is shown at the right of the chamber.

Fig. 2 shows the interior of the test chamber, which is 10 ft in diameter by 30 ft long. In the background is a test setup for carburetor de-icing tests, while on the left is a test jig for determining the coefficient of expansion of an aircraft control cable. In the upper right of the picture are shown the cold air ducts with deflectors and the electric unit heaters which are connected to the automatic temperature control.

Cork insulation is installed inside the chamber in order to reduce the heat load given off by the steel shell when the chamber air is suddenly cooled down. This cork is 6 in. thick and installed without any adhesives or fasteners by using the principle of an arch. Insulation, 2 in. thick, is installed on the outside of the shell. The upper shell, which holds the cooling coils, is insulated in the same manner.

Fig. 3 shows the main control station. The large oval port in front of the operator permits observation of a large part of the test chamber. Push but-

tons for the refrigeration equipment and vacuum pumps are located on this panel. The three air operated controllers control the rate of climb, temperature, and pressure automatically. The operator is in constant communication with persons inside the chamber with the man lock operator and the medical observer. Other sets of instruments which apply to the equipment being tested are usually located to the right of this main control station.

It is sometimes necessary for persons to enter or leave the main test chamber while it is at altitude conditions. In order to do this they must first enter the man lock, which is a small chamber attached to one end of the main chamber. If they enter from the main chamber, then the man lock pressure

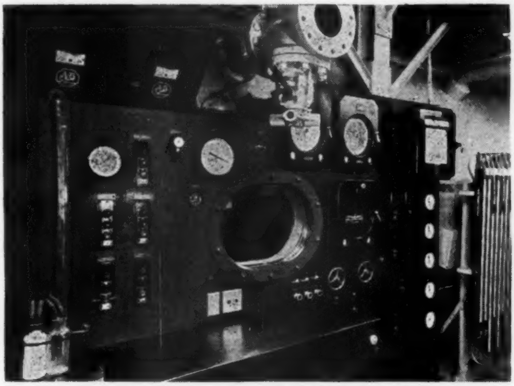

FIG. 3. CONTROL STATION OUTSIDE OF ALTITUDE CHAMBER

is first made equal to the chamber pressure. After entering the man lock, the men close the door through which they have just passed and then the man lock pressure altitude is lowered. When the pressure equals atmospheric pressure, the outside door may be opened and the men step out. If simulated altitudes are high enough during this operation the men carry portable walk-around oxygen bottles when transferring from the main chamber to the man lock.

Fig. 4 shows the control station for the man lock. The operator who controls the vacuum and bleed in air manually is in communication with the persons inside the chamber as well as with the main control station operator and the medical doctor.

The refrigeration equipment consists essentially of a three stage compression system using Freon-12 as a refrigerant. It develops a total of 36 tons of refrigeration with an evaporator temperature of —85 F. The rotary vacuum pumps for creating simulated high altitude total 150 hp and they can be arranged for parallel single stage operation or for series two stage operation.

With all of the foregoing equipment, performance becomes quite remarkable. Secrecy forbids publication of the maximum altitude or the minimum tempera-

ture which this altitude chamber can produce, but a brief statement of the requirements of one of the acceptance tests for this equipment will give a good idea of its performance. There was to be installed in the chamber during such a run the fuselage of a pursuit airplane plus its mechanical equipment which gave off a specified heat load. It was required that the altitude in the chamber be raised from ground level to 35,000 ft and at the same time that

FIG. 4. CONTROL STATION FOR MAN LOCK

the temperature be dropped from +60 F to −70 F in a period of nine minutes. Just imagine an airplane trying to match that performance in climbing to high altitude.

EFFECT OF ALTITUDE ON MAN

Let's review briefly some of the problems encountered by men and machines at high altitude. Some of these will be more or less familiar to heating and air conditioning engineers; others will be entirely new.

First of all, let's take the human being. Man is definitely a ground animal and he can only adapt himself to reasonable changes from his natural environment.

Oxygen Requirement

Man requires additional oxygen if he goes above 10,000 ft for any length of time, and even when breathing 100 per cent pure oxygen, he cannot live at an altitude much above 40,000 ft without special means, because the atmospheric pressure at that altitude is too low to force sufficient oxygen into the blood stream at the lungs. In order to exist at an altitude much higher than 40,000 ft, the flier must be surrounded by a local atmospheric pressure sufficient to force the oxygen into the blood stream. This is accomplished by using pressurized cabins, pressurized suits and other pressuring means. Two Army Air Corps men, Stevens and Anderson, have reached an altitude of 72,395 ft in a balloon fitted with a spherical gondola which was pressurized and air conditioned.

Temperature Requirement

Another change of which man can tolerate very little is that of temperature. He must be supplied with added heat or clothing if his skin temperature drops much below 88 F and must have external cooling if his skin temperature rises much above this figure. Clothing is used as an insulation against changes of temperature. This gets very bulky if it is sufficient to insulate for temperatures as low as −70 F, which are often met in the stratosphere. (Actually temperatures of −110 F have been found in the stratosphere.)

Aeroembolism

Aeroembolism is another factor which makes it discomforting and even hazardous for man to climb to high altitudes. Aeroembolism is an illness similar to the diver's *bends* and generally occurs only above 28,000 ft altitude. It is generally considered to be caused by the gases which are normally entrained in the blood stream and tissues coming out of solution and forming bubbles. These bubbles often collect near the joints and under the skin and become either annoying or extremely painful, depending on the severity of the condition. The only known remedy at present is to increase the pressure of the atmosphere around the person in order to put those gases back into solution. This is done either by pressurization such as with pressure cabins or by descending to a lower altitude where the pressure is greater.

Almost everybody is familiar with ear trouble which occurs with rapid changes of altitude or pressure. The sinuses are similarly affected by these changes. Another result of pressure change is the expansion of abdominal gases. If these cannot be readily relieved, they become pocketed and, as the pressure surrounding a person is lowered, the gas in the pocket expands and causes distension of the bowels which if carried very far becomes painful.

Dry Oxygen and Air

Fliers who have breathed oxygen for a long time such as on long bombing missions are often aware of a raw throat and nostrils due to the breathing

of bone dry oxygen. Oxygen has to be bone dry so that it won't freeze in the regulators. Not much experience has yet been accumulated on pressure cabins for airplanes, but it is predicted that dehydration of the personnel in these cabins may be a serious problem when flying at high altitudes unless some arrangement is made to humidify the air. At +70 F and sea level saturated air has about 8.00 grains of moisture per cubic foot, but at −60 F and 33,000 ft altitude it has only 0.01 grain per cubic foot. At lower temperatures than this, the amount of moisture is nil. Since the cabin supercharger must take this very dry air from the surrounding atmosphere to pressurize

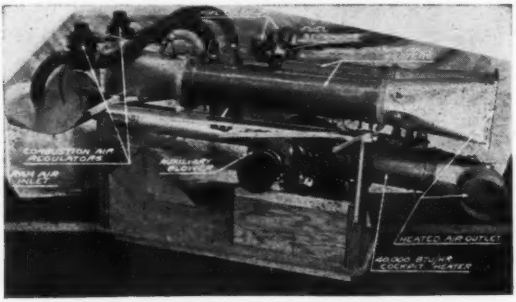

Fig. 5. Gasoline Fired Airplane Heater

the cabin and make up for ventilation and leaks, it is evident that a real problem is created.

Other factors such as acceleration, effect of noxious gases, use of drugs and a few others should also be included in the list but are not discussed in this paper.

Effect of Altitude on Equipment

High altitude problems are not limited entirely to physiology of human beings. Man made machinery is even more troublesome. Most materials become very brittle at cold temperatures. Ordinary rubber hose cracks when flexed and some metals become brittle as glass and shatter on impact.

Expansion and Contraction

The difference in the rate of contraction or expansion of materials also presents a serious problem because of the wide temperature ranges encountered and the close fits of many parts, such as in gun and turret mechanisms and the hundreds of other precision parts which go into an airplane. The ambient temperatures encountered may vary from as high as 120 F on the ground to —100 F in the stratosphere. Under such conditions, close fitting parts made of materials with dissimilar coefficients of expansion may seize.

Viscosity of Oils

The change in viscosity of oils and other fluids presents a serious problem in aircraft where the ambient temperature may change from +100 F on the ground to −70 F in the stratosphere in a matter of a few minutes. This fact has led to considerable research in the development of hydraulic fluids and greases for aircraft.

Electrical Equipment

Electrical equipment at high altitudes is likewise troublesome. Due to low pressures, sparking and corona discharge are more severe and require heavy insulation of various parts such as spark plug cables or even pressurization of them. Bad radio interference results if this is not remedied.

Excessive wear of brushes on d-c generators has been found at high altitudes. The major cause is generally conceded to be lack of humidity in the air. Moisture or moisture combined with other things is a lubricant. Without it excessive friction and wear will occur.

MISCELLANEOUS PROBLEMS

There are many other problems too numerous to discuss in detail but some of these should be mentioned. They are: icing and fogging of windshields and windows; devising suitable controls and seals for pressurized cabins and methods of supercharging engines and pressure cabins; and trouble with communication systems, particularly microphones.

The studies of carburetor icing, coefficient of expansion of materials, and of many physiological problems not yet mentioned have also been successfully carried out in altitude chambers.

HEATING AIRPLANES

Some of the problems encountered in high altitude flight are more or less familiar to heating and ventilating engineers. Airplane heating is one of these. Off-hand it may look like a fairly simple job to install a heater and some ducts in an airplane and thereby obtain satisfactory heat. One of the first considerations confronting the designer is that of weight. Aircraft engineers are well aware of the saying *$100 a pound.* It means that for every pound of dead weight saved, the earning capacity of the airplane is increased $100 during its lifetime. Some airline operators think $200 a pound is nearer the correct value.

The constant struggle is to achieve the optimum balance between weight and comfort. Where you try to save weight by making the ducts smaller and selecting the smallest heater possible, sometimes you find that the smallest heater is the least efficient or has the least output so you try to counteract that by insulating the cabin. Immediately up goes the weight again.

At high altitudes, the skin of the airplane gets very cold; in fact, nearly as cold as the outside air. This brings the *cold wall effect* into the selection of comfort conditions. Radiation from a man's body to the cold wall is quite appreciable and insulation must again be considered. So important is this

cold wall effect that cabin temperatures taken during acceptance tests for aircraft heating systems are sometimes obtained by installing the thermocouples in small aluminum blocks painted dull black. This method will give a reading lower than the actual air temperature but will be a better indication of the degree of warmth felt by an individual inside that cabin.

The very fact that the atmosphere is less dense at high altitudes makes it difficult to heat the cabin by convection. Pressurizing the cabin would of course eliminate this particular problem.

While it is not within the scope of this paper to discuss in detail the various types of heaters being used, two popular types will be mentioned: the gasoline

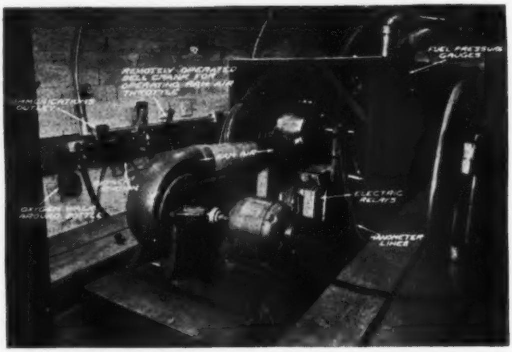

FIG. 6. TEST ARRANGEMENT FOR AIRPLANE HEATER

fired heater and the exhaust gas heat exchanger. The gasoline fired *heater*, Fig. 5, burns regular aviation gasoline and is very light and compact, enabling it to be tucked away in small spaces near the spot to be heated. Fig. 6 shows this same heater undergoing tests in an altitude chamber. This unit consists of three separate heaters totaling 240,000 Btu per hr. The altitude chamber can be very helpful as an aid to the development of aircraft combustion heaters. Tests in the chamber proved that the heaters would burn without supercharged combustion air at altitudes higher than any regular military or commercial plane has flown. They also revealed many weaknesses in the accessory equipment which would normally be discovered only after months of actual service.

The *exhaust gas heat exchanger* takes waste heat from the engine exhausts and distributes the heated air in ducts to the points of application. This device requires no extra fuel, is simple to operate and has an ample amount of heat available during most flight conditions. Objections to this system have been primarily the added weight of duct work necessary and the failure of the heat exchangers due to corrosion and vibration. If a plane is originally designed for this type of system, the added weight of the ducts can be much

less than if an existing plane has to be modified. This system is used successfully in some German planes.

PRESSURIZED CABINS

Mention was made earlier in this paper of pressurized cabins for aircraft. While these eliminate many physiological problems, they do add others both physiological and mechanical. The question usually asked when pressure cabins for military aircraft are mentioned is *What happens when a shell hits it?* This problem was one which had to be solved before the B-29 Super Fortresses could be put into combat operation. The sudden reduction of cabin pressure resulting from the puncturing of a pressure cabin is termed *explosive decompression.* Although the details on *explosive decompression* cannot be revealed for security reasons, the author can say from personal experience that such an encounter is not fatal. Most of the work on this physiological problem has been done by the Army Air Forces Aero-Medical Laboratory in altitude chambers.

Much remains to be done in the field of passenger comfort at high altitudes and it is going to take engineers who are familiar with both airplane design and heating and air conditioning to do a good job. The altitude chamber will be a very useful instrument for aiding in the solution of this problem and others caused by high altitude.

DISCUSSION

W. E. ZIEBER, York, Pa. (WRITTEN): The author is to be commended for the apt manner in which he has shown the many problems facing us from the aviation standpoint in the war effort.

My associates have been faced with a few of these problems since they have furnished much apparatus of types similar to that described but they could not know the extent of the problems that the author in his position would be compelled to solve.

Because of the time element involved in the war effort, designers are forced to use their past knowledge and experience to make such apparatus operate as well as possible today. Time and continued effort will reveal knowledge for improvement.

In this apparatus, in order to accomplish the speed of reduction of temperature required in the lower shell, refrigeration is stored in the mass of the upper shell before attempting to cool the lower one. Insulation is put on the inside of the shells to prevent trouble with the shell material. No insulation adhesive is used because none has been found that would hold at low pressures.

More could be said about such problems to show that those who build such apparatus have many things to learn. It is possible that the methods of producing low temperatures can be improved in the future.

We, as engineers, will accept the responsibilities outlined by the author. We will need to cooperate in every possible way to accomplish the requirements of the war.

The many needs which will arise after the war are going to demand much attention. In order to be able to take care of the comfort and safety of human beings and maintain the proper performance of equipment and instruments the closest cooperation will be required between the air conditioning engineers and the engineers of the aircraft manufacturers and operators. This cooperation will be necessary in the research phases as well as those of application. It will be necessary in times of peace to obtain knowledge for use when and if there are any future wars.

H. B. NOTTAGE, East Hartford, Conn. (WRITTEN): The most valuable feature of this paper is the challenge which is offered to one's imagination in considering the

sort of quantitative data and methods which would be involved in establishing solutions to the important aircraft problems mentioned.

The author has emphasized the vital importance of *weight* as an aircraft design factor. In the usual designs for residences and buildings, weight enters only indirectly into the cost of space occupied and represented by so much material. But the foundations and structure of buildings are designed according to principles which *in no way approach* the *extremely critical* balance of the many component elements of an airplane which must fly through the skies as a single compact unit. For instance, it has been estimated [1] that each pound added to an accessory of a pursuit-class aircraft will result in an overall increase of 5 lb to the entire take-off weight of the airplane! The airplane must be designed as a *matched assembly* of many parts to meet an exacting specified performance.

Many Society members are interested in combustion and perhaps feel that they are identified with highly effective types of combustion equipment. However, the problems of maintaining efficient combustion in aircraft heaters with high-altitude air have only served to emphasize the unsolved problems which abound in the combustion field. It seems appropriate in this connection to ask if the group at Curtiss-Wright Research has accomplished anything in this particular regard.

Then, too, the physiological problems of maintaining human comfort in flight are much more critical than for ground structures. If a person feels uncomfortable while in a building, no vital danger is ordinarily presented to him thereby. But, in the case of military personnel or crew members, discomfort may be a prelude to acute difficulties which might seriously endanger either the individual or the entire aircraft. Quick action in the response of a person to an emergency will be hampered by a lack of oxygen or by serious overcooling of portions of the body. Also, commercial travelers pay for *comfort*. The author has mentioned the importance of *radiation* in the comfort heat balance, and he has indicated the further difficulties of corrective equalization of air temperature with rarified air under cabin conditions.

These problems, and others of equipment design, installation, and operation would seem to belong in the progressively expanding domain of heating, ventilating, and air conditioning. All of the many problems which come up for altitude chamber study must ultimately yield to expression in terms of engineering experience and physical law. It is to be hoped that the Society will continue to actively contribute its proper share in this field, for there is much to be done.

M. K. FAHNESTOCK, Urbana, Ill.: I think it is very timely and opportune to bring this matter of the air conditioning of aircraft to the attention of the Society, and the only remark that I have to make is to de-emphasize military aviation and point out that post-war aviation as a commercial enterprise is going to be an entirely different thing than military aviation. The only remark made by the previous speakers having to do with commercial aviation involved economics, which of necessity is entirely neglected by the military. It was the question of the weight of heaters. Of course, we realize that weight is important to all aviation. It is worth from $100 to $150 per pound per year on a DC-3 in commercial aviation.

Then there is this so-called matter of comfort. We are not concerned too much with comfort in military aviation, because discomfort usually comes before a fellow falls down physiologically from the standpoint of performance. So there is a difference in that in military aviation we are concerned with performance more than with comfort; while in civilian aviation we are concerned with comfort, because that is a part of what we are selling in air transportation, and air transportation is in competition with other means of transportation.

There were several other remarks, about high altitude and rates of climb, both of which are of course important problems in military aviation, but of much less importance in civilian aviation. When you are talking about these 30,000, 35,000, 40,000

[1] The Intercooling Problem in Airplane Design by H. B. Dickinson (*SAE Journal*, Vol. 52, p. 557, December, 1944).

and 45,000 altitudes, you are talking about ships that will not be in commercial aviation for a long, long time. The hazards of explosive decompression and things of that sort are almost entirely problems of military aviation. There are other factors that have not been mentioned which come into civilian aviation that I think might be emphasized. One of them is odor. We are concerned with that in military aviation, particularly in evacuation hospital ships, but in normal military aviation, in such planes as the B-29, and the B-17, odors as such are no problem.

I merely make these remarks to point out that there is a great difference between military aviation where economics are not considered at all and civil aviation where it is of utmost importance. In warfare the job must be done the best way regardless of cost. However, in civilian aviation, which is in competition with other means of transportation, the problems are almost entirely different. They are certainly different from the economic standpoint.

As a society serving a new field of transportation it would be well to recognize the problems of civilian aviation in addition to those of military aviation. That follows in regard to refrigeration decompression chambers and other laboratory apparatus for test purposes because the extremes of atmospheric environment involved are somewhat different in the two aviation fields.

E. A. Norman, Jr., Columbus, Ohio: I did not come here with the intention of speaking to you, but my company happens to be one that builds aircraft heaters. We did not jump into this business overnight, as Mr. Crowell has pointed out. We have been building these highly specialized heaters for several years now and the complications are enormous. There are so many problems that come up that normal heating men just cannot realize what they are. We are in the throes at the moment of building a high-altitude test chamber. We have availed ourselves of the chamber at Curtiss-Wright many times.

I would like to take exception to Mr. Nottage's statement that military aircraft are not comfort heated. Anyone who has experienced temperatures of 30 F or lower and has then flown in a plane where the inside temperature is 30 F or more has comfort heating as far as he is concerned. It must be remembered that our fighting planes were designed within certain weight and space limitations and that heat was almost an afterthought. Consequently any distribution system not incorporated in the original planes cannot hope to do what we would regard as a true comfort heating job. With commercial transport planes the problem is somewhat simplified because the value of correct heating is realized and designers are prepared to give up the space required for a good system. The field is wide open for further development.

In obtaining the maximum Btu in the minimum of space, we are now at a point where we can burn at the rate of 5 to 6 million Btu per cubic foot of combustion space. That requires enormous velocities of air to take that heat away. I think before the year is out we will be up to 10 million Btu's but that is for war usage and would never be practical for peacetime aircraft.

The only additional thing I want to say is that, in this business, we welcome anybody as well as contributions and ideas. We are trying to do the best job we can but there is a lot more to be learned about this business than we know at the present.

L. T. Avery, Cleveland, Ohio: There is one point Mr. Crowell didn't have time to emphasize that should be emphasized as it shows how closely the air conditioning problem ties in with refrigeration and also with physiological reactions. I refer you to the paragraph on Dry Oxygen and Air (see p. 117). Prof. L. E. Seeley pointed out to me that our constitution calls us an Association for studying the arts and the science of heating, ventilating and air conditioning, but nothing has been said about physiology. I do not think we need the title changed, but we cannot avoid the responsibility of the interrelationship.

Chairman Winslow: I think adequate data are available to estimate the seriousness of that problem from Professor Seeley's work and from the work that we have done at the Pierce Laboratory. It can very easily be determined just how great will

be the loss of moisture from the upper respiratory tract since this loss is equivalent to the effect of altering the absolute amount of moisture in the air breathed in and out from its original value to the value of approximately 90 per cent saturation at 95 deg.

AUTHOR'S CLOSURE: We have been talking about heating the airplane all this time. Aircraft speeds are getting higher and higher and, as I mentioned in my discussion, the boundary layer air temperature is higher than ambient air temperature due to the fact that kinetic energy is converted into heat when the air is slowed down in the boundary layer next to the airplane skin. This boundary layer temperature increases as the square of the velocity, and at 600 mph amounts to about 60 F increase. With 100 F ambient air temperature this would mean 160 F airplane skin temperature. We are now at the point where we have to cool the airplane. I think some consideration should be given to the cooling system.

1270

No. 1270

THERMODYNAMIC PROPERTIES OF MOIST AIR

By John A. Goff * and S. Gratch,** Philadelphia, Pa.

This paper is the report of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the Towne Scientific School, University of Pennsylvania.

Introduction

The ultimate objective of this cooperative investigation has been a formulation of the thermodynamic properties of moist air which, on account of accuracy and thermodynamic consistency, could claim universal acceptance as standard.

The results of the investigation are presented in Table 4, Thermodynamic Properties of Moist Air (which is intended as a revision of the present Table 6, Chapter 1, in the 1944 and 1945, Heating, Ventilating, Air Conditioning Guides) and in the accompanying detailed explanation of the method used in constructing Table 4. The explanation as presented should offer a means of judging the reliability and probable permanence of the data.

It is hoped that the methods employed in constructing the tables will be helpful and instructive to those who have occasion to deal with the general problem of gas mixtures.

I. Dry Air

Composition. For the purpose of analysis, moist air can be regarded as a mixture of only two constituents, dry air and water vapor. This is so because the composition of dry air is practically constant both as to time and location, the only detectable variation being in the carbon dioxide (CO_2) content. According to *International Critical Tables,*[1] the mol-fraction composition of dry air is that given in Table 1. All monatomic gases except argon (A) are present in traces only; hence, in computing their contributions to apparent molecular weight, enthalpy, and entropy, no appreciable error is committed by lumping them together as argon.

Table 1 also lists the molecular weights used in the present analysis. Those for CO_2 and H_2 are from *International Critical Tables,*[2] those for N_2 and A are from Cragoe,[3] that for O_2 is a matter of definition. With these individual molecular weights the apparent molecular weight of the mixture, dry air, is readily computed.

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[1] Exponents refer to Bibliography at end of paper.

Presented at the 51st Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, Mass., January, 1945.

Specific Volume. At all accessible temperatures, say above 0.001 °R, the zero-pressure value of the pressure-volume product pv (ft lb/mol) is strictly proportional to absolute temperature T (°R), the constant of proportionality being the universal gas constant R (ft lb/mol °R). According to Cragoe,[3] the best value of the product RT_o, where T_o denotes ice-point temperature, is 22414.6 (atm cm³/gmol). Beattie[4] recommends the value $T_o = 273.165 \pm 0.015$ °K. Combining these and converting to English units yield $R = 1545.31$ (ft lb/mol °R).

TABLE 1—COMPOSITION OF DRY AIR

Gas	Mol-fraction	Molecular Weight	lb/mol dry air
Nitrogen (N_2)	0.7803	28.0149	21.8600
Oxygen (O_2)	0.2099	32.0000	6.7168
Carbon Dioxide (CO_2)	0.0003	44.003	0.0132
Hydrogen (H_2)	0.0001	2.016	0.0002
Monatomic Gases	0.0094	39.943	0.3755

Apparent Molecular Weight (lb/mol)...28.966

Composition of Monatomic Gases

Argon (A)	0.00933
Neon (Ne)	0.000018
Helium (He)	0.000005
Krypton (Kr)	0.000001
Xenon (Xe)	0.———
	0.00935

At sufficiently low, but nevertheless finite pressure p, the specific volume is given accurately by the expression,

$$pv = RT - A_{aa}(T)\cdot p - A_{aaa}(T)\cdot p^2 \qquad 2.0$$

where the temperature functions $A_{aa}(T)$ and $A_{aaa}(T)$ are called second and third virial coefficients, respectively. The letter a refers specifically to dry air; the meanings of the double and triple subscripts will be explained later.

Numerical values of second and third virial coefficients are derivable from Bridgeman's[5] adjustment of the constants in the well-known Beattie-Bridgeman equation of state[6] to fit the experimental data of Holborn and Schultze,[6] namely,

$$\left.\begin{aligned} R &= 0.08206 \\ T_o &= 273.13 \\ A_o &= 1.0763 \\ B_o &= 0.04070 \\ a &= 0.01697 \\ b &= -\,0.02174 \\ c &= 12 \times 10^4 \end{aligned}\right\} \qquad 2.1$$

The units of these constants are such that each term in the following equations obtained from rearrangement of the Beattie-Bridgeman equation is

dimensionless if pressure is expressed in atmospheres and temperature in degrees Kelvin.

$$\frac{A_{aa}p}{RT_o} = -\frac{B_o p}{RT_o} + \frac{A_o p/R^2T_o}{T} + \frac{cp/RT_o}{T^3} \quad \ldots \quad 2.2$$

$$\frac{A_{aaa}p^2}{RT_o} = \frac{B_o(B_o + b)p^2/RT_o}{T} - \frac{A_o(2B_o + a)p^2/R^3T_o}{T^2} + \frac{A_o^2p^2/R^4T_o}{T^3}$$

$$- \frac{cB_op^2/R^2T_o}{T^4} + \frac{2cA_op^2/R^3T_o}{T^5} + \frac{c^2p^2/R^2T_o}{T^7} \quad \ldots \quad 2.3$$

After inserting Bridgeman's values of the constants, the following data are easily computed for a pressure of 1 atmosphere:

t (F)	176	−160
RT/RT	1.000000	1.000000
$-A_{aa}p/RT$	0.000029	−0.004691
$-A_{aaa}p^2/RT$	0.000003	−0.000012
pv/RT	1.000032	0.995297

These calculations show that, at both extremes of the temperature range of present interest, the contribution to the specific volume from the third virial coefficient A_{aaa} is less than the probable uncertainty in the contribution from the second virial coefficient A_{aa}, at any rate at atmospheric pressure.

Converting (2.2) to English units with temperature expressed in degrees, Rankine gives

$$A_{aa}(T) = -0.6520 + \frac{378.18}{T} + \frac{11.2 \times 10^6}{T^3} \text{ (ft}^3\text{/mol)} \quad \ldots \quad 2.4$$

This equation is purely empirical and involves adjustment of three constants. Goff, Andersen and Gratch [7] have shown that fair agreement with (2.4) is obtained by the adjustment of only one constant in a semi-theoretical expression for the force potential $E(r)$ appearing in the statistical mechanical expression for the second virial coefficient of a non-polar gas, namely,

$$A(T) = 2\pi N_o \int_0^\infty (e^{-E(r)/kT} - 1)r^2dr \quad \ldots \quad 2.5$$

N_o = Avogadro's number
k = Boltzmann's constant
r = distance between members of a pair of molecules.

The forces contributing to the potential E are of two sorts, long-range attractive forces and short-range repulsive forces. In the case of a non-polar gas, the only long-range attractive forces are the so-called *dispersion forces* arising from momentary deformation of the electron clouds surrounding the atoms which, in an aggregate of molecules, produces a tendency toward alignment into attractive orientations. The contribution to the potential E from such forces is a negative one depending only on distance of separation r and being of the form

$$E_-(r) = -(a_3r^{-6} + a_4r^{-8} + a_5r^{-10}) \quad \ldots \quad 2.6$$

The coefficients a_3, a_4, a_5 are derivable from optical dispersion data.

As previously noted, dry air is not a pure gas so that (2.5) is not strictly applicable to it. However, all of its constituents are non-polar; and its two prin-

cipal constituents, oxygen (O_2) and nitrogen (N_2), show roughly the same values of the coefficients a_3, a_4, a_5. Therefore, it ought to be a good approximation to the truth to regard dry air as a pure gas, assigning to it dispersion force coefficients obtained by averaging those of its principal constituents weighted in proportion to their mol fractions in the mixture. The values thus obtained for dry air from data by Margenau[7, 8] are:

$$\left.\begin{aligned} a_3 &= 53.6 \times 10^{-60}\ (\text{erg cm}^6) \\ a_4 &= 114.9 \times 10^{-76}\ (\text{erg cm}^8) \\ a_5 &= 127.9 \times 10^{-92}\ (\text{erg cm}^{10}) \end{aligned}\right\} \qquad 2.7$$

At very short distances of separation the force between a pair of molecules becomes repulsive, the potential E becoming positively infinite as r decreases

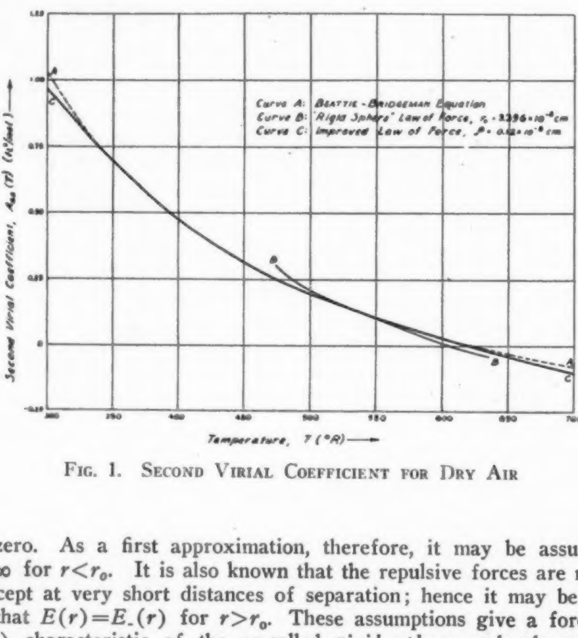

FIG. 1. SECOND VIRIAL COEFFICIENT FOR DRY AIR

toward zero. As a first approximation, therefore, it may be assumed that $E(r) = \infty$ for $r < r_0$. It is also known that the repulsive forces are negligibly small except at very short distances of separation; hence it may be assumed further that $E(r) = E_-(r)$ for $r > r_0$. These assumptions give a force potential $E(r)$ characteristic of the so-called ***rigid sphere*** molecule which has been used successfully in several cases.[9] With it the integral in (2.5) is relatively easy to evaluate. In the preliminary analyses referred to,[7] the value $r_0 = 3.396 \times 10^{-8}$ cm was found to bring (2.5) into fair agreement with (2.4) within a limited temperature range as shown by curves A and B, Fig. 1.

From the foregoing comparison it is clear that the ***rigid sphere*** model is hardly adequate for the purpose at hand. A better method of treating the repulsive forces is to assume their contribution to $E(r)$ except at extremely short distances of separation (say $r < 0.5 \times 10^{-8}$ cm) to be of the form $Pe^{-r/\rho}$, where P and ρ are constants. This form with $\rho = 0.28 \times 10^{-8}$ cm is known to be ap-

propriate for some of the simplest molecules [10, 11] and, as will be shown, it enables (2.5) to give satisfactory agreement with (2.4) in the case of dry air. The improved law of force becomes

$$E(r) = \infty \text{ for } r < 0.5 \times 10^{-8} \text{ cm} \qquad 2.8$$
$$E(r) = Pe^{-r/\rho} - (a_6 r^{-6} + a_8 r^{-8} + a_{10} r^{-10}) \text{ for } r \geqq 0.5 \times 10^{-8} \text{ cm.}$$

It would appear from (2.8) that there are now two constants to be adjusted, but it turns out that ρ may be assigned *a priori* any value in the range 0.10×10^{-8} to 0.30×10^{-8} cm and the effect of such assignment almost completely compensated, except at very high temperatures, by subsequent adjustment of P. The values $\rho = 0.12 \times 10^{-8}$ cm, $P = 5.056 \times 10^{-2}$ erg give curve C of Fig. 1.

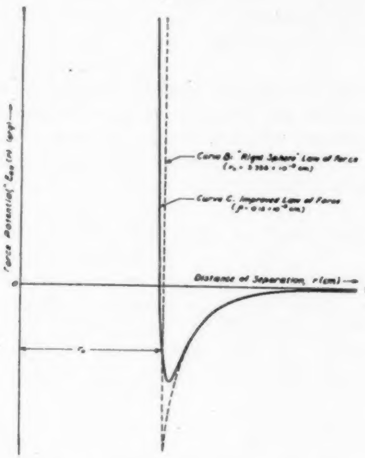

FIG. 2. LAWS OF FORCE FOR DRY AIR

The corresponding force potential $E(r)$ is sketched in Fig. 2, which also shows a dotted line representing the *rigid sphere* potential used in calculating curve B of Fig. 1. It is not contended that these values represent the best possible adjustment, but merely that they prove the adequacy of the theory which will be put to important use later.

Enthalpy and Entropy. Expressions for specific enthalpy and specific entropy are derivable from (2.0) by application of appropriate identical relations of thermodynamics. The results are:

$$h_a = h_a^\circ(T) - B_{aa}(T)\,p - \tfrac{1}{2} B_{aaa}(T)\,p^2 \qquad 3.0$$

$$s_a + R \log_e p = s_a^\circ(T) + C_{aa}(T)p + \tfrac{1}{2} C_{aaa}(T)p^2 \qquad 3.1$$

where

$$B = d(A\tau)/d\tau \qquad 3.2$$
$$C = dA/dT \qquad 3.3$$
τ = reciprocal temperature

Using Bridgeman's constants (2.1) the following data are easily computed for a pressure of 1 atmosphere

t (° F)	176	−160
$B_{aa}p/RT$	+0.001535	+0.016165
$\frac{1}{2}\, B_{aaa}p^2/RT$	−0.000003	+0.000071
$C_{aa}p/R$	−0.001564	−0.011475
$\frac{1}{2}\, C_{aaa}p^2/R$	+0.000001	−0.000065

These calculations show that, at both extremes of the temperature range of interest, the contributions to specific enthalpy and specific entropy from the third virial coefficient A_{aaa} are less than the probable uncertainties in the corresponding contributions from the second virial coefficient A_{aa}, at any rate at atmospheric pressure. Consequently, in subsequent calculations at pressures not greatly exceeding atmospheric, the third virial coefficient A_{aaa} will be completely ignored.

From (3.0) it is clear that $h_a°(T)$ is the zero-pressure enthalpy. Until recent developments in the science of interpreting band spectra made it possible to calculate $h_a°(T)$ from spectroscopic data, the only method of evaluating it was to extrapolate to zero pressure experimental measurements of h_a made at accessible pressures. At present, however, the calculated values are generally regarded as possessing a much higher degree of accuracy and reliability than the extrapolated values.

Analysis of the spectrum of a given molecule determines a set of discrete energy levels ϵ_i ($i = 0, 1, 2, \ldots.$). From quantum mechanical theory it is known that it is often possible for more than one accessible state to have the same energy level ϵ_i; hence it is also necessary to know the statistical weight p_i of each level, that is, the number of accessible states having that level. Given this information, it is a straightforward, though sometimes quite complicated, task to compute the so-called state sum defined as follows:

$$Q = \sum_{i=0}^{\infty} p_i e^{-\epsilon_i/kT} \quad \ldots \ldots \quad 3.4$$

But statistical mechanics predicts that the zero-pressure enthalpy is derivable from the state sum Q as follows:

$$\frac{h° - \bar{u}}{RT} = T\left[\frac{\partial \log_e(QT)}{\partial T}\right]_v \quad \ldots \ldots \quad 3.5$$

The constant $\bar{u}$ is the internal energy at zero temperature and zero pressure and is called the null-point energy. Because of the fact that, even at absolute zero ($T = 0$), there persists appreciable translational kinetic energy in a gas, there is a finite difference between null-point energy and null-point enthalpy. However, this difference is extremely small except in the case of an electron gas and can therefore be ignored in the present analysis. In problems not involving chemical changes the null-point energy can arbitrarily be assigned any convenient value.

Dry air is a mixture of several gases, but statistical mechanics predicts that the zero-pressure enthalpy of such a mixture is the sum of separate contributions from its several constituents; thus

$$h_a° = \Sigma\, x_i\, (h_i° - \bar{u}_i) + \Sigma\, x_i \bar{u}_i \quad \ldots \ldots \quad 3.6$$

where x denotes mol fraction. In constructing Table 4, the constant $\Sigma\, x_i \bar{u}_i$ has been adjusted to make the specific enthalpy h_a zero at 0 F and standard atmospheric pressure.

For nitrogen (N_2) we have computed the state sum Q using the same spectroscopic data employed by Johnston and Davis.[12] These data do not include values of the energy levels ϵ_i (erg) but rather those of wave numbers ν_i(cm^{-1}). These are related in a simple manner, namely, $\epsilon_i = hc\nu_i$, *where* h denotes Planck's constant (erg sec) and c the velocity of light *in vacuo* (cm/sec). In the state sum Q the various physical constants appear only in the combination hc/k(cm °K), and for this combination the value 1.4324 was used. Even though more accurate spectroscopic data and more accurate values of the physical constants than those employed by Johnston and Davis may now be available, it did not seem necessary to search them out, because, in the temperature range of interest, the corrections determined by them would certainly be entirely negligible.

For oxygen (O_2) the state sum calculations of Johnston and Walker[13] were used directly, semi-theoretical equations being employed in the necessary interpolation to the temperatures listed in Table 4. Since the mol fraction of oxygen in dry air is only about 21 per cent, inaccuracies in the Johnston and Walker values are not reflected into the final results.

For the remaining constituents of dry air (Table 1) values in the literature[14] were accepted without question since the concentrations are sufficiently small to submerge any possible inaccuracies.

From (3.1) it is clear that $s_a°(T)$ is the zero-pressure value of the sum, $s_a + R \log_e p$, which may be called *reduced entropy*. The zero-pressure reduced entropy of a pure gas is derivable from the state sum Q as follows:

$$\frac{s° - \bar{s}}{R} = \frac{h° - \bar{u}}{RT} + \log_e \left(\frac{QRT}{ev}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 3.7$$

Actually Q is proportional to specific volume v, the ratio Q/v being dimensionless. This makes $s°/R$ a pure temperature function as required by (3.1). In computing the logarithm in (3.7) the unit of R must be so chosen that the quantity RT/v has the unit desired for pressure.

The zero-pressure reduced entropies of the several constituents of dry air were obtained from the same sources as were the zero-pressure enthalpies. For the mixture, namely, dry air, statistical mechanics predicts that

$$s_a° = \Sigma\, x_i\,(s_i° - \bar{s}_i) - R\,\Sigma\, x_i \log_e x_i + \Sigma\, x_i \bar{s}_i \quad \ldots\ldots\ldots\ldots \quad 3.8$$

Although statistical mechanics further predicts (Third Law of Thermodynamics) that, even when chemical changes are contemplated, each of the constants $\bar{s}_i$ may be assigned the value zero, this procedure has not been followed. Instead the constant $\Sigma\, x_i \bar{s}_i$ has been adjusted to make the specific entropy s_a zero at 0 F and standard atmospheric pressure.

Summary. The values of specific volume of dry air v_a listed in the proposed revision of Table 6 have been computed from (2.0), using $R = 1545.31$ (ft lb/mol °R) and $M = 28.966$ (lb/mol), obtaining A_{aa} from (2.2) after

inserting the Bridgeman constants (2.1), and ignoring the third virial coefficient A_{aaa}. The values of specific enthalpy h_a have been computed from (3.0), obtaining B_{aa} from (2.2) through (3.2), ignoring the third virial coefficient B_{aaa} constructing h_a° in accordance with (3.6) from individual zero-pressure enthalpies computed from spectroscopic data with $R = 1.9858$ Btu/mol °R and from the mol fractions listed in Table 1, adjusting the constant $\Sigma\ x_i\bar{u}_i$ to make $h_a = 0$ at 0 F and standard atmospheric pressure. The values of specific entropy s_a have been computed from (3.1), obtaining C_{aa} from (22) through (3.3), ignoring C_{aaa}, constructing s_a° in accordance with (3.8) from individual zero-pressure reduced entropies computed from spectroscopic data, adjusting the constant $\Sigma\ x_i\bar{s}_i$ to make $s_a = 0$ at 0 F and standard atmospheric pressure.

II. WATER

Introduction. Since publication in 1936 of the Keenan-Keyes steam tables,[15] new experiments and theoretical developments have indicated that the data included are inaccurate at low pressures and temperatures. The *Bureau of Standards* calorimetric measurements [16] published in 1939 indicate an error of about 0.7 Btu/lb in the steam table value of the enthalpy of saturated steam at 32 F. Measurements of isothermal Joule-Thomson coefficient B_{ww} reported by Collins and Keyes [17] in 1938 show that the values of this coefficient used in formulating the steam tables are in error by about 15 per cent in the range 40 to 60 C, and by about 10 per cent at 80 C. Recalculation of the zero-pressure properties by Stephenson and McMahon [18] from the newer spectroscopic data of Randall, Dennison, Ginsberg, and Weber [19] indicate an error of as much as 0.4 Btu/lb in zero-pressure enthalpy in the range 0 to 100 C. The steam table values of vapor pressure in this range are not as accurate as desired, the tolerance of 0.05 per cent at 80 C set by the Third International Steam Table Conference [20] in 1935 producing an uncertainty of 0.1 per cent in the humidity ratio of saturated moist air at atmospheric pressure and a corresponding uncertainty of 0.5 Btu/lb dry air in its enthalpy. If the thermodynamic properties of moist air were to be computed from present steam table data, the overall uncertainty would be too large to warrant their acceptance as standard; for example, the overall uncertainty in the enthalpy of saturated moist air would probably exceed 1 Btu/lb of dry air.

If the aforementioned uncertainties in the properties of water could only be reduced by additional experimentation, which would necessarily have to be quite elaborate, the end might not justify the means. Fortunately, however, a considerable reduction can be accomplished by correlation of the newer data already available. We have attempted such correlation and shall describe it briefly here with the intention of publishing a full account later. For the sake of completeness the properties of condensed water (liquid and solid phases) are included.

Condensed Water. It is universal practice in steam table work to select saturated liquid at 32 F as reference point, assigning the value zero to both enthalpy and entropy at that point. This practice has been adhered to in constructing Table 4. Osborne [21] has recalculated earlier data on the latent heat of fusion at 32 F and atmospheric pressure and has recommended the value

333.48±0.2 int. joule/gram. To correct this figure to saturation pressure, the identical relation of thermodynamics, namely,

$$\left(\frac{\partial h}{\partial p}\right)_T = \left[\frac{\partial(v\tau)}{\partial \tau}\right]_p \quad \ldots\ldots \quad 6.1$$

may be used, evaluating the right hand member (1) for ice from data on thermal coefficient of expansion by Jakob and Erk [22] and the value of specific volume at 32 F given by Keenan and Keyes [15] and (2) for liquid from the data of Osborne, Stimson and Ginnings.[16] The correction to the enthalpy turns out to be only 0.001 Btu/lb and therefore negligible. Accordingly the enthalpy and entropy of ice at 32 F and saturation pressure are −143.40 Btu/lb and −0.29164 Btu/lb °F, respectively. Values of the enthalpy and entropy of ice at lower temperatures can now be obtained from the calorimetric measurements of Giauque and Stout [23] after correction from atmospheric to saturation pressure and conversion to the steam table Btu.

For the properties of liquid water at saturation pressure and above 32 F, the *Bureau of Standards* [16] values are used directly.

Zero-pressure Properties of Water Vapor. Gordon [24] has calculated the zero-pressure properties of water vapor from the spectroscopic data of Mecke et al.[25] Wilson [26] has subsequently estimated a correction to the Gordon values to account for the centrifugal distortion of the water molecule, the so-called *stretching effect.* Stephenson and McMahon [18] have evaluated this effect more accurately from the newer spectroscopic data of Randall, Dennison, Ginsberg, and Weber.[19] We have found that the best information presently available is adequately † represented, in the range 100 to 1000 K by the following semi-theoretical formulae:

$$\frac{h^\circ - \bar{u}}{T} = 0.440912 + 0.110228 \sum_{i=1}^{3} \left(\frac{\theta_i\tau}{e^{\theta_i\tau} - 1} + a_i\,\theta_i\tau e^{-\theta_i\tau}\right)$$

$$+ 1.427 \times 10^{-6}\, T - \frac{0.9943}{T}\ (\text{Btu/lb}\,^\circ\text{R}) \quad \ldots\ldots \quad 7.1$$

$$s^\circ - \bar{s} = 0.440912 \log_e T + 0.110228 \sum_{i=1}^{3} \left[\frac{\theta_i\tau}{e^{\theta_i\tau} - 1} + \log_e\left(\frac{e^{\theta_i\tau}}{e^{\theta_i\tau} - 1}\right)\right.$$

$$\left. + a_i\,(1 + \theta_i\,\tau)\, e^{\theta_i\tau}\right] + 2.854 \times 10^{-6}\, T - 0.26958\ (\text{Btu/lb}\,^\circ\text{R}) \ldots \quad 7.2$$

In these formulae the *characteristic temperatures* θ_i and the coefficients a_i have been obtained by least squares adjustment to best fit the data described previously; temperature is to be expressed in degrees Rankine (°R); the unit chosen for pressure in $s^\circ = (s + R \log_e p)_{p=0}$ is the standard atmosphere; and the molecular weight of water is 18.0154 lb/mol. The numerical values of the constants are:

$$\theta_1 = 4124.09\ ^\circ\text{R} \qquad a_1 = -0.03958$$
$$\theta_2 = 9317.47\ ^\circ\text{R} \qquad a_2 = 0.05353$$
$$\theta_3 = 9802.06\ ^\circ\text{R} \qquad a_3 = 0.04000$$

† For the range 298.1 to 1500°K: root mean square deviation = 0.00003 for (7.1), 0.00002 for (7.2); mean algebraic deviation = +0.000006 for (7.1), −0.000002 for (7.2).

The null-point constants $\bar{u}$ and $\bar{s}$ remain to be fixed in such a way that the enthalpy and entropy of saturated liquid at 32 F will both be zero in accordance with steam table practice.

Virial Coefficients. Empirical formulae for second and third virial coefficients are derivable from the formulation of Keyes, Smith and Gerry [27] based on their own measurements of specific volume at temperatures in the range 195 to 460 C. These are:

$$A_{ww} = -1.89 + 2641.62\,\tau \cdot 10^{80870\tau^2}\ (\text{cm}^3/\text{g}) \quad \ldots \quad 8.1$$

$$A_{www} = (A_{ww}\tau)^2\,(-82.546 + 1.6246 \times 10^5\tau)\ (\text{cm}^3/\text{g atm}) \quad \ldots \quad 8.2$$

where temperature is in degrees Kelvin (°K).

These formulae have been used for the necessary extrapolation to lower temperatures in constructing the Keenan-Keyes tables.[15] Subsequently, Collins and Keyes [17] found that values of B_{ww} derived from (8.1) are in error by about 15 per cent at 40 C and about 3 per cent at 125 C. They also found that values of B_{www} derived from (8.2) are in error by about 20 per cent at 80 C and about 15 per cent at 125 C.

The Collins and Keyes [17] data cover the range 40 to 125 C only and are not sufficiently accurate or extensive to stand extrapolation to the lower temperatures in Table 4. Furthermore, their measurements were made directly on B_{ww} and B_{www} so that only the temperature dependence, not the absolute values, of A_{ww} and A_{www} are derivable therefrom. The Keyes, Smith, Gerry [27] formulation (8.1), (8.2) cannot be expected to yield reliable values of these constants of integration since extrapolation from 195 to 125 C would be required. Fortunately, however, the new *Bureau of Standards* [16] calorimetric measurements furnish a way out of the dilemma.

The *Bureau of Standards* [16] data include values of saturation enthalpy h_g of high precision and may be used to evaluate the left hand side of

$$B_{ww}\,p_s + \tfrac{1}{2}\,B_{www}\,p_s^2 = (h^\circ - \bar{u})_w - h_g + \bar{u}_w \quad \ldots \quad 8.3$$

to within a constant quantity $\bar{u}_w$. This can be fixed by comparison with the values given by Collins and Keyes [17] for B_{ww} at 38.94, 59.44, 80.02, 100, and 125 C and for B_{www} at 80.02, 100, and 125 C, using existing vapor pressure data p_s which are sufficiently accurate for the purpose. The value thus obtained is 859.114 Btu/lb with an average deviation of only 0.01 Btu/lb which may be taken to indicate a high degree of consistency in the various data used. The necessary extrapolation of B_{www} to temperatures below 80 C to enable all values of B_{ww} to be used in fixing $\bar{u}_w$ can hardly introduce appreciable uncertainty into the adjustment of $\bar{u}_w$, because the term $\frac{1}{2}\ B_{www}\ p_s^2$ decreases so rapidly with decreasing temperature and amounts to less than 0.1 Btu/lb at 59.44 C.

Having fixed the constant $\bar{u}_w$ in the foregoing manner values of the sum, $B_{ww}\ p_s + \frac{1}{2} B_{www}\, p_s^2$, were computed from (8.3) over the range 0 to 125 C; then values of $\frac{1}{2}\ B_{www}\ p_s^2$ computed from the data of Collins and Keyes [17] and suitably extrapolated to *OC* were subtracted to obtain values of $B_{ww}\ p_s$. These were divided by existing values of vapor pressure p_s and the resulting values of B_{ww}, together with the direct measurements of Collins and Keyes,[17] were then formulated into the empirical equation (8.5). Finally, the values of

$\frac{1}{2}\, B_{www}\, p_s^2$ used in the reduction were divided by $\frac{1}{2}\, p_s^2$ to obtain corresponding values of B_{www} which in turn were formulated (somewhat arbitrarily, since only three values were available for the purpose) into the empirical equation (8.8). Using the identical relations (3.2) and (3.3), the remaining temperature coefficients are readily obtained, the complete list being

$$A_{ww\tau} = -\,0.0304\,\tau + 90.432\tau^2 e^{530{,}020\tau^2} + C_0 \qquad 8.4$$

$$B_{ww\tau} = -\,0.0304\tau + (A_{ww\tau} + 0.0304\tau - C_0)\,(2 + 1{,}060{,}640\tau^2) \qquad 8.5$$

$$C_{ww} = A_{ww\tau} - B_{ww\tau} \qquad 8.6$$

$$A_{www\tau} = 66.1\,(A_{ww\tau} - C_0)^3 + C_1 \qquad 8.7$$

$$B_{www\tau} = 198.3\,(A_{ww\tau} - C_0)^2\, B_{ww\tau} \qquad 8.8$$

$$C_{www} = A_{www\tau} - B_{www\tau} \qquad 8.9$$

where the first three are given in ft³/lb °R and the last three in ft⁶/lb² °R.

To fix the constants of integration C_0 and C_1 values of pv at saturation for temperatures above 95 C, where sufficiently reliable values of vapor pressure and its temperature derivative are obtainable from the formula of Osborne and Meyers,[28] were computed from the identical relation,

$$(pv)_s = \gamma\,\frac{d\log_e T}{d\log_e p_s} \qquad 8.10$$

where γ is a calorimetric quantity directly measured at the *Bureau of Standards*,[16] and compared with values computed for the same temperatures from

$$(pv)_s = RT - A_{ww}p_s - A_{www}p_s^2 \qquad 8.11$$

It was found in this manner that both C_0 and C_1 were zero within the accuracy of the data used.

The above empirical formulae may be regarded as valid from 0 to 125 C. They do not merge smoothly with the Keyes, Smith and Gerry[27] formulation valid above 195 C, but this is not considered a serious defect so long as they are not to be used above 100 C.

In the previous section on dry air, considerable discussion of the application of modern theory regarding intermolecular forces to predict values of second virial coefficient was given. In the case of water vapor, the problem is much more complex due to the polar character of the water molecule which makes the potential E depend upon mutual orientation as well as on distance of separation. We have found that the constants P and ρ together with certain fictitious constants expressing the effects of dipole alignment and induction can be adjusted to fit existing data on B_{ww} although such adjustment does not yield unique values of A_{ww} as was originally hoped. It is interesting to note, however, that the adjusted value of ρ was near 0.24×10^{-8} cm. More recently Margenau and Myers[29] have refined the theory to take proper account of the polar character of the water molecule and have obtained fair agreement with data on A_{ww}.

Vapor Pressure. As mentioned previously, the values of vapor pressure recommended by the Third International Steam Table Conference[20] are not as reliable in the low temperature range as desired for the construction of standard moist air tables. Thus the assigned tolerances are 0.05 per cent at

90 C and 0.10 per cent at 0 C. But the *Bureau of Standards*[19] measurements of γ have an uncertainty probably less than 0.01 per cent from 0 to 100 C which fact suggests the possibility of calculating the vapor pressure from (8.10) and (8.11) using the expressions (8.4) and (8.7) (with $C_0 = C_1 = O$) for the virial coefficients in (8.11).

Details of this calculation will be omitted, but the final expression for the temperature derivative of $\log_{10} p_s$ is given as follows:

$$\frac{d \log_{10} p_s}{dT} = \frac{5308.90 + 1720\, e^{-.01234T} + 1.6945\, e^{11.5776 - 5682.0T}}{T^2} - \frac{2.18362}{T} + 4.2 \times 10^{-6}\, e^{-0.05\,(t-32)} + f(t) \quad \ldots \; (T \text{ in } ^\circ\text{R}) \qquad 9.1$$

where the residual $f(t)$ is best handled numerically, being very small. This expression can be integrated downward from 212 F where the vapor pressure

TABLE 2—VALUES OF f(t)

t (deg F)	$10^7 f(t)$	$10^6 \int_{212}^{t} f(t)\, dt$
32	−2	46
50	−4	40
68	−6	32
86	−6	21
104	−6	11
122	−4	2
140	−2	−5
158	0	−7
176	2	−6
194	2	−2
212	0	0

is exactly one atmosphere by definition to obtain values of $\log_{10} p_s$ where p_s is expressed in atmospheres. However, it serves no important purpose to write out the resulting expression so the matter will be left with a tabulation of the residual $f(t)$ and its integral as given in Table 2.

It is believed that the uncertainty in values of $\log_{10} p_s$ calculated from (9.1) and Table 2 is not greater than 0.01 per cent. All values are well within the tolerances set by the Third International Steam Table Conference.[20]

Vapor Pressure of Ice. Goff[30] has recently calculated the vapor pressure of ice from the Clapeyron relation

$$T \frac{dp_s}{dT} = \frac{h_g - h_i}{v_g - v_i} \qquad 10.1$$

The authors have repeated these calculations using the present information regarding virial coefficients. Integration of (10.1) gives values of $\log_{10}$ (p_s/p_0), where p_0 denotes the vapor pressure at 32 F, in substantial agreement with the previously published results.

According to *International Critical Tables,*[31] the triple point where the vapor pressure of ice is identical with that of the liquid, is at 32.0176 F. At 32 F

the vapor pressure of the liquid is 0.0060281 atm according to the integral of (9.1). From two writings of the Clapeyron relation, one for equilibrium between vapor and ice and one for equilibrium between vapor and liquid, the difference between the vapor pressure of liquid and that of ice at 32 F can be estimated. The answer is 0.0000006 atm; accordingly the value assigned to p_0 is 0.0060275 atm.

It is believed that the uncertainty in the recalculated values of $\log_{10} (p_s/p_0)$ is about 0.02 per cent. This is equivalent to an uncertainty in p_s itself of about 0.13 per cent at −40 F and about 0.34 per cent at −160 F. Harrison [32] has shown that the previously calculated values of p_s agree very well with existing experimental data.

Other Steam Properties. Since $\bar{u}_w$ has been determined and the virial coefficients B_{ww} and B_{www} formulated, values of saturation enthalpy of the vapor h_g can be calculated from (8.3) in the range 32 to 212 F. Dividing the value thus obtained for 32 F by the absolute temperature gives a value of saturation entropy s_g to be used in fixing the constant $\bar{s}_w$ in

$$s_g = -R \log_e p_s + (s^\circ_w - \bar{s}_w) + C_{ww}p_s + \tfrac{1}{2} C_{www}p_s^2 + \bar{s}_w \quad . \; . \; . \; . \; . \quad 11.1$$

after which values of s_g can be computed at other temperatures in the foregoing range.

Values of saturation volume v_g are to be obtained from (8.11).

Summary. Values of second and third virial coefficients for water have been distilled from (1) zero-pressure enthalpies obtained from a critical review of existing state-sum calculations based on spectroscopic data, (2) saturation enthalpies measured calorimetrically and with high precision at the *Bureau of Standards,* (3) isothermal Joule-Thomson coefficients measured by Collins and Keyes, (4) values of vapor pressure above 95 C from Osborne and Meyers, (5) values of the calorimetric quantity γ measured at the *Bureau of Standards,* (6) the best value of the ratio of the absolute to the international kilowatt-hour as recently recommended by the *Bureau of Standards* (1.00019).

From these data the vapor pressure of liquid water was calculated for the range 32 to 212 F with an estimated uncertainty of only 0.01 per cent in the logarithm of the vapor pressure (in atmospheres) over the whole range. The vapor pressure of ice has been recalculated from additional data on (1) latent heat of fusion at 32 F and atmospheric pressure reported by Osborne, (2) thermal coefficient of expansion by Jakob and Erk, (3) the specific volume at 32 F from Keenan and Keyes, (4) specific heat at atmospheric pressure from Giauque and Stout. The range of the recalculations is −160 to 32 F with an estimated uncertainty of only 0.02 per cent in the logarithm of the ratio of the vapor pressure to that at the triple point.

Finally, other steam properties have been calculated and will be proposed as a revision of existing steam tables in the low temperature range.

III. MOIST AIR

Theory of Gas Mixtures. Statistical mechanics predicts that at sufficiently low pressure p the specific volume of a gas mixture is given as to form by the equation,

$$pv = RT - p \sum_{ik} x_i x_k A_{ik}(T) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad 13.1$$

where x_i denotes mol fraction of constituent i, x_k denotes mol fraction of constituent k, $A_{ik}(T)$ denotes second virial coefficient if $i=k$, interaction coefficient if i is unequal to k. It is clear that the coefficients $A_{ik}(T)$ express the effects of interaction between *pairs* of molecules, *like* pairs if $i=k$ and *unlike* pairs if i is unequal to k, and that the sum extends over all pairs with the understanding, of course, that $A_{ik}(T) = A_{ki}(T)$.

If (13.1) be applied to a pure gas by putting all mol fractions except number one equal to zero and putting number one equal to unity, the result is

$$pv = RT - A_{11}(T)\,p \qquad 13.2$$

Unfortunately, the discussion of the preceding section has shown that (13.2) cannot accurately represent the properties of saturated water vapor at temperatures much above 150 F where a term in pressure squared of the form $-A_{111}(T)p^2$ must be added. By analogy it may be supposed that $A_{111}(T)$ expresses the effect of interaction between members of a group of three like molecules; it is called the third virial coefficient.

We know of no valid information, either theoretical or experimental, as to how (13.1) should be modified to include the effect of interaction between three-member groups of molecules, but have assumed by analogy (but nevertheless quite arbitrarily) that a proper modification is

$$pv = RT - p\sum_{ik} x_i x_k A_{ik}(T) - p^2 \sum_{ijk} x_i x_j x_k A_{ijk}(T) \qquad 13.3$$

Corresponding expressions for specific enthalpy h, specific entropy s, and the chemical potentials μ_i are obtained from (13.3) by application of appropriate identical relations of thermodynamics together with predictions of statistical mechanics regarding the additivity of zero-pressure properties as follows:

$$h = \sum_i x_i h_i^\circ - p\sum_{ik} x_i x_k B_{ik} - \tfrac{1}{2} p^2 \sum_{ijk} x_i x_j x_k B_{ijk} \qquad 13.4$$

$$s + R\log_e p = \sum_i x_i s_i^\circ - R\sum_i x_i \log_e x_i + p\sum_{ik} x_i x_k C_{ik} + \tfrac{1}{2} p^2 \sum_{ijk} x_i x_j x_k C_{ijk} \qquad 13.5$$

$$\mu_i = g_i^\circ(T) + RT\log_e(x_i p) + p\Big(\sum_{ik} x_i x_k A_{ik} - 2\sum_k x_k A_{ik}\Big) + \tfrac{1}{2} p^2 \Big(2\sum_{ijk} x_i x_j x_k A_{ijk} - 3\sum_{jk} x_j x_k A_{ijk}\Big) \qquad 13.6$$

where as before

$$B = d(A\tau)/d\tau \qquad 3.2$$

$$C = dA/dT \qquad 3.3$$

and where $g_i^\circ(T)$ is the zero-pressure reduced specific free enthalpy, namely,

$$g_i^\circ = h_i^\circ - T s_i^\circ \qquad 13.7$$

To use (13.3) to (13.6) in calculating properties of moist air would require information on the coefficients A_{aaa}, A_{aaw}, A_{aww}, A_{www}. As shown in Section I, the first of these, together with its derivatives B_{aaa} and C_{aaa}, are negligible in the range of temperature and pressure considered. Suitable values of the last, together with its derivatives B_{www} and C_{www}, have been deduced in Sec-

tion II. But there exists no information, either experimental or theoretical, regarding A_{aaw}, A_{aww}, and their respective derivatives. This means that our calculated properties of moist air above 150 F must be offered with some reservations as to accuracy and permanence. What we have done is to neglect the coefficients A_{aaw} and A_{aww} entirely and write

$$pv = RT - [x^2 A_{aa} + x(1-x)\, 2\, A_{aw} + (1-x)^2 A_{ww}]\, p - [(1-x)^3 A_{www}]\, p^2 \qquad 13.8$$

$$h = [xh_a^\circ + (1-x)\, h_w^\circ] - [x^2 B_{aa} + x(1-x)\, 2\, B_{aw} + (1-x)^2 B_{ww}]\, p - \tfrac{1}{2}[(1-x)^3 B_{www}]\, p^2 \qquad 13.9$$

$$s + R \log_e p = [xs_a^\circ + (1-x)\, s_w^\circ] - R\,[x \log_e x + (1-x)\log_e (1-x)] + [x^2 C_{aa} + x(1-x)\, 2\, C_{aw} + (1-x)^2 C_{ww}]\, p + \tfrac{1}{2}[(1-x)^3 C_{www}]\, p^2 \qquad 13.10$$

$$\mu_a = g_a^\circ + RT \log_e (xp) + [(A_{ww} - 2\, A_{aw} + A_{ww})(1-x)^2 - A_{aa}]\, p + [(1-x)^3 A_{www}]\, p^2 \qquad 13.11$$

$$\mu_w = g_w^\circ + RT \log_e (1-x)_p + [A_{ww} - 2\, A_{aw} + A_{ww})\, x^2 - A_{ww}]\, p - [(1-x)^2 (\tfrac{1}{2} + x)\, A_{www}]\, p^2 \qquad 13.12$$

where x denotes the mol fraction of dry air and $(1-x)$ that of water vapor.

Saturation. When two distinct phases, say liquid and vapor or solid and vapor, coexist, each is said to be saturated with respect to the other. From general thermodynamic reasoning the conditions for saturation are found to be equality between the two coexisting phases of (1) pressure, (2) temperature, and (3) each component chemical potential. Expressions for the chemical potentials of dry air and water in the vapor phase are given by (13.11) and (13.12), respectively. It remains, therefore, to obtain suitable expressions for these component chemical potentials in the condensed phase.

Actually the condensed phase in the present problem is almost pure water but containing a very small amount of dissolved (and oxygen enriched) air. For the purpose of analysis, the condensed phase can be regarded as an ideal, incompressible solution whose composition can be specified by x', the mol fraction of dry air. For such a solution the component chemical potentials are given as to form by

$$\mu_a' = RT \log_e x' + \overline{V}_a'\, p + g_a'(T) \qquad 14.1$$

$$\mu_w' = RT \log_e (1-x') + \overline{V}_w'\, p + g_w'(T) \qquad 14.2$$

where $\overline{V}_a'$ and $\overline{V}_w'$ denote component partial molal volumes, and the prime refers to the condensed phase.

One condition for saturation is obtained by equating (13.11) and (14.1), and it may be reduced to the form,

$$x_a' = k'(T)\, x_a\, p \qquad 14.3$$

where, strictly speaking, $k'(T)$ should depend on pressure as well as temperature though empirically its pressure dependence is found to be extremely slight and is, therefore, neglected in the following analysis.

Another condition for saturation is obtained by equating (13.12) and (14.2) and replacing $\log_e(1-x_s')$ by $-x_s'$ which is justified by the smallness of x_s' compared with unity. The result is

$$\begin{aligned} RT \log_e[(1 - x_s)\, p] = & - RT\, k'\,(T)\, x'_s\, p + V'_w\, p \\ & - [(A_{aa} - 2\, A_{aw} + A_{ww})x_s^2 - A_{ww}]\, p \\ & + [(1 - x_s)^2\, (\tfrac{1}{2} + x_s)\, A_{www}]\, p^2 \\ & + [g_w'\,(T) - g_w^\circ\,(T)] \quad \ldots\ldots\ldots\ldots \quad 14.4 \end{aligned}$$

The additional conditions of equality, between the two coexisting phases, of pressure p and temperature T, have been applied in deriving (14.4) itself.

Now the temperature functions $g_w'\,(T)$ and $g_w^\circ\,(T)$ can be eliminated in favor of vapor pressure p_s (of pure water) by noting that for $p = p_s$, $x_s = x_s' = 0$. Thus, (14.4) becomes

$$\begin{aligned} RT \log_e \left[\frac{(1 - x_s)\, p}{p_s}\right] = & - RT\, k'\, x_s p + \bar{V}_w'\,(p - p_s) \\ & - [(A_{aa} - 2\, A_{aw} + A_{ww})\, x_s^2]\, p + A_{ww}\,(p - p_s) \\ & + [A_{www}\,(x_s - 3/2)x_s^2]\, p^2 + \tfrac{1}{2}\, A_{www}\,(p^2 - p_s^2) \\ & \ldots\ldots\ldots\ldots\ldots\ldots \quad 14.5 \end{aligned}$$

and determines mol fraction x_s as a function of pressure p and temperature T, since p_s is itself a known temperature function. Since x_s appears on both sides, a trial-by-error calculation would be required to get an arbitrarily precise solution (14.5), but such calculations clearly indicate that a sufficiently precise solution is obtained by substituting $(1 - p_s/p)$ for x_s in the right-hand side. Such substitution and subsequent division by RT reduce (14.5) to

$$\begin{aligned} \log_e \left[\frac{(1 - x_s)\, p}{p_s}\right] = & \left(\frac{\bar{V}_w' p_s}{RT} - k'\, p_s\right)\left(\frac{p}{p_s} - 1\right) \\ & + \left(\frac{2A_{aw}\, p_s}{RT} - \frac{A_{aa}\, p_s}{RT}\right)\left(\frac{p}{p_s} - 2 + \frac{p_s}{p}\right) \\ & + \left(\frac{A_{ww}\, p_s}{RT} + \frac{A_{www}\, p_s^2}{RT}\right)\left(1 - \frac{p_s}{p}\right) \quad \ldots\ldots \quad 14.6 \end{aligned}$$

The Coefficients $\bar{V}_w'$ *and* $k'(T)$. At saturation the condensed phase is almost pure water. Therefore, the partial molal volume $\bar{V}_w'$ does not differ appreciably from the specific volume v_f (or v_i) of pure saturated liquid (or solid) water. Values of the solubility coefficient $k'(T)$ for the liquid phase have been taken from *International Critical Tables* [33]; those for the solid phase have been assumed zero since the solubility of air in ice is negligibly small.

The Interaction Coefficient A_{aw}. A considerable portion of the cooperative investigation referred to in the introduction to this paper was devoted to an experimental measurement of the interaction coefficient A_{aw}. The experiment devised for the purpose has been called the two-stage, saturation-isotherm experiment and has been described in a previous paper.[34] In it a continuous sample of dry, carbon dioxide-free air was passed through a suitable saturator from which it left at a closely regulated pressure p_1, thence through a reducing valve, thence through a drying train in which m_1 pounds of water picked up in the saturator was removed (during the test period), thence through a second

saturator from which it left at a closely regulated pressure p_2, thence through a second drying train in which m_2 pounds of water was removed (during the same period), thence to the suction of a vacuum pump discharging to atmosphere. Both saturators were located in a single thermostated bath so that the air was saturated twice at the same temperature T, but at two different pressures.

If the weight of dry air m_a passing through the apparatus during the test period had been measured with sufficiently high precision, the mol fractions x_1 and x_2 could have been calculated and corresponding values of A_{aw} computed from two writings of (14.6), one for saturation at pressure p_1 and the other for pressure p_2. These two computed values should be identical since A_{aw} is supposed to depend on temperature only, but actually the common temperature was not measured with sufficient accuracy to enable this method to yield reliable results. Alternatively, m_a was eliminated between the two writings of (14.6), leaving A_{aw} as the only unknown and making the value computed in this manner practically unaffected by slight inaccuracies in the measurements of temperature and weight of dry air.

The results obtained in the above manner but with less reliable values of the properties of dry air and water vapor have been published previously.[7] Recalculated values are shown in Table 3.

TABLE 3—INTERACTION COEFFICIENT FOR MOIST AIR

t (deg C)	$\lambda = \dfrac{2A_{aw}}{A_{aa} + A_{ww}}$	A_{aw} (ft³/mol)
5	0.048	0.64 ± 0.03
15	0.054	0.61 ± 0.03
20	0.060	0.61 ± 0.03
25	0.059	0.54 ± 0.03

In a previous paper[7] it was shown that surprisingly close agreement with the experimental values of A_{aw} was obtained from the theoretical expression (2.5) using the *rigid sphere* model for expressing $E_{aw}(r)$ in terms of distance of separation r, choosing for r_0 the arithmetic mean of the corresponding r_0-values found for dry air and water vapor separately as suggested by Fowler,[35] and determining the coefficients a_3, a_4, a_5 in (2.6) from corresponding coefficients for dry air and water vapor by means of combination rules suggested by the theory. Fortunately, (2.5) is applicable to the calculation of A_{aw} even though it is not to A_{ww}, because the interaction potential E_{aw} is not appreciably affected by the polar character of the water molecule and can therefore be regarded as a function of distance of separation only.

The close agreement referred to encouraged us to improve upon the rigid sphere model in order to use the theory for the necessary extrapolation of A_{aw} well outside the experimental range of temperature. The improvement consisted of expressing the contribution to $E_{aw}(r)$ from short-range repulsive forces in the form $P_{aw}e^{-r/\rho_{aw}}$, assigning ρt_{aw} the value 0.28×10^{-8} cm *a priori*, determining the coefficients a_3, a_4, a_5 from the data of Margenau,[1] and adjust-

ing P_{aw} to fit the experimental data in Table 3. The various constants used are listed as follows:

$$\left.\begin{aligned} P_{aw} &= 3.4337 \times 10^{-9}\ \text{erg} \\ \rho_{aw} &= 0.28 \times 10^{-8}\ \text{cm} \\ a_3 &= 54.7 \times 10^{-60}\ \text{erg cm}^6 \\ a_4 &= 114.9 \times 10^{-76}\ \text{erg cm}^8 \\ a_5 &= 127.9 \times 10^{-92}\ \text{erg cm}^{10} \end{aligned}\right\} \qquad 16.1$$

These constants place the potential minimum very near 3.230×10^{-8} cm, the value of r_0 obtained for the rigid sphere model in the manner explained.

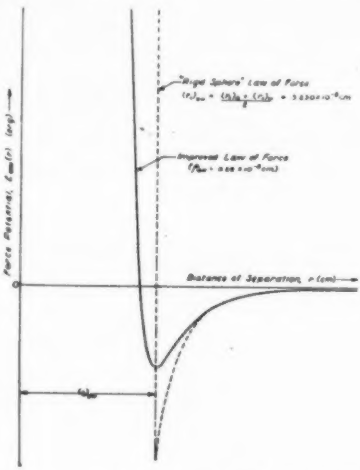

FIG. 3. LAWS OF FORCE FOR AIR-WATER INTERACTION

Fig. 3 shows the two assumed laws of force. Fig. 4 shows the values of A_{aw} calculated from the improved law, and it is felt that they deserve considerable confidence, especially since they are obtained by the adjustment of only one constant in a theory which has already registered many successes. We have formulated these values of A_{aw} into the following empirical equation:

$$A_{aw} = -0.4730 + 5.950 \times 10^{-5}\, T\left(1 - e^{-\frac{7949.7}{T}}\right) + \frac{505.91}{T} + \frac{4949}{T^2} + \frac{7.957 \times 10^6}{T^3}\ (\text{ft}^3/\text{mol}) \qquad 16.2$$

where T is °R.

The bold extrapolation represented in Fig. 4 requires additional discussion. In the first place, some sort of extrapolation is necessary because, during war conditions, it became impossible to extend the range of experimental values

beyond what had already been covered. That the particular method used deserves considerable confidence is chiefly because it involves the adjustment of only one constant in a well-tested theory. It is true, of course, that the theory involves several approximations so that the possibility that the extrapolated values of A_{aw} may differ appreciably from the true values cannot be denied. Still the effect of A_{aw} on the various properties of moist air happens to be greatest right in the experimental range and is almost negligible at both extremes of the extrapolated range. Thus, while the estimated uncertainty of 5 per cent in A_{aw} at 68 F produces an uncertainty of 0.015 per cent in the

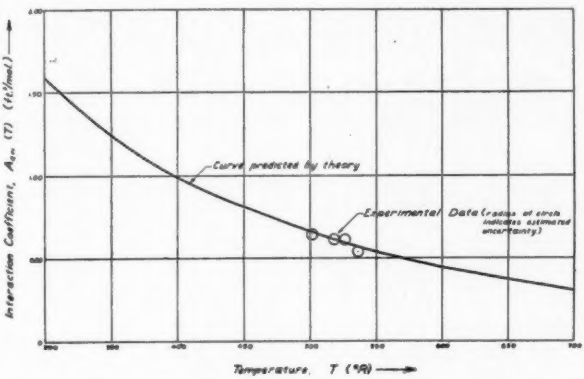

FIG. 4. EXTRAPOLATION OF EXPERIMENTAL DATA ON INTERACTION COEFFICIENT

humidity ratio W_s of saturated moist air (at atmospheric pressure), the same uncertainty in W_s at 192 F would require an uncertainty of 70 per cent in A_{aw} there. Also, while the effect of A_{aw} on W_s is greater at the very low temperatures, its effect on the other thermodynamic properties is negligible because W_s is itself so very small. Thus an error of 20 per cent in A_{aw} at -160 F produces an error of 0.3 per cent in W_s, but of only 0.00000001 Btu/lb dry air in h_s, the enthalpy of saturated moist air (at atmospheric pressure); furthermore, an uncertainty 0.3 per cent in W_s is less than that produced by the present uncertainty of the vapor pressure data at -160 F.

IV. TABLE 4

The first item to be entered in Table 4, Thermodynamic Properties of Moist Air at Standard Atmospheric Pressure, is the vapor pressure p_s. For temperatures below and at 32 F, the values listed are those for equilibrium over ice recalculated as explained under the heading, *Vapor Pressure of Ice.* Strictly speaking, these should be extended upward to 32.0176 F, the triple point where solid, liquid, and vapor phases coexist. There is a double entry at 32 F, the second representing an extrapolated value of the vapor pressure

TABLE 4—THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^6$	Volume cu ft/lb dry air v_a	v_{as}	v_s	Enthalpy Btu/lb dry air h_a	h_{as}	h_s	Entropy Btu per (°F) (lb dry air) s_a	s_{as}	s_s	Condensed Water Enthalpy Btu/Lb h'_w	Entropy Btu/(Lb)(°F) s'_w	Vap. Press. In. Hg $p_s \times 10^6$	Fahr. Temp. t(F)
−160	0.2120	7.520	0.000	7.520	−38.504	0.000	−38.504	−0.10300	0.00000	−0.10300	−222.00	−0.4907	0.1009	−160
−159	0.2394	7.545	0.000	7.545	−38.262	0.000	−38.262	−0.10219	0.00000	−0.10219	−221.68	−0.4896	0.1139	−159
−158	0.2703	7.571	0.000	7.571	−38.021	0.000	−38.021	−0.10139	0.00000	−0.10139	−221.36	−0.4885	0.1286	−158
−157	0.3049	7.596	0.000	7.596	−37.779	0.000	−37.779	−0.10059	0.00000	−0.10059	−221.04	−0.4874	0.1450	−157
−156	0.3435	7.622	0.000	7.622	−37.538	0.000	−37.538	−0.09980	0.00000	−0.09980	−220.72	−0.4864	0.1635	−156
−155	0.3869	7.647	0.000	7.647	−37.296	0.000	−37.296	−0.09901	0.00000	−0.09901	−220.40	−0.4853	0.1842	−155
−154	0.4354	7.673	0.000	7.673	−37.055	0.000	−37.055	−0.09822	0.00000	−0.09822	−220.07	−0.4843	0.2073	−154
−153	0.4897	7.699	0.000	7.699	−36.813	0.000	−36.813	−0.09743	0.00000	−0.09743	−219.75	−0.4832	0.2331	−153
−152	0.5502	7.724	0.000	7.724	−36.572	0.000	−36.572	−0.09664	0.00000	−0.09664	−219.42	−0.4822	0.2620	−152
−151	0.6178	7.750	0.000	7.750	−36.330	0.000	−36.330	−0.09586	0.00000	−0.09586	−219.10	−0.4811	0.2942	−151
−150	0.6932	7.775	0.000	7.775	−36.088	0.000	−36.088	−0.09508	0.00000	−0.09508	−218.77	−0.4800	0.3301	−150
−149	0.7772	7.801	0.000	7.801	−35.847	0.000	−35.847	−0.09430	0.00000	−0.09430	−218.44	−0.4789	0.3701	−149
−148	0.8709	7.826	0.000	7.826	−35.606	0.000	−35.606	−0.09352	0.00000	−0.09352	−218.11	−0.4779	0.4146	−148
−147	0.9750	7.851	0.000	7.851	−35.364	0.000	−35.364	−0.09274	0.00000	−0.09274	−217.78	−0.4768	0.4641	−147
−146	1.091	7.876	0.000	7.876	−35.123	0.000	−35.123	−0.09198	0.00000	−0.09198	−217.45	−0.4758	0.5194	−146
−145	1.219	7.902	0.000	7.902	−34.881	0.000	−34.881	−0.09121	0.00000	−0.09121	−217.12	−0.4747	0.5807	−145
−144	1.362	7.927	0.000	7.927	−34.640	0.000	−34.640	−0.09044	0.00000	−0.09044	−216.78	−0.4737	0.6488	−144
−143	1.521	7.953	0.000	7.953	−34.399	0.000	−34.399	−0.08967	0.00000	−0.08967	−216.45	−0.4726	0.7243	−143
−142	1.698	7.978	0.000	7.978	−34.157	0.000	−34.157	−0.08892	0.00000	−0.08892	−216.11	−0.4716	0.8082	−142
−141	1.893	8.004	0.000	8.004	−33.916	0.000	−33.916	−0.08816	0.00000	−0.08816	−215.78	−0.4705	0.9011	−141
−140	2.109	8.029	0.000	8.029	−33.674	0.000	−33.674	−0.08740	0.00000	−0.08740	−215.44	−0.4695	1.004	−140
−139	2.348	8.054	0.000	8.054	−33.433	0.000	−33.433	−0.08664	0.00000	−0.08664	−215.11	−0.4684	1.118	−139
−138	2.613	8.079	0.000	8.079	−33.192	0.000	−33.192	−0.08589	0.00000	−0.08589	−214.77	−0.4674	1.244	−138
−137	2.906	8.105	0.000	8.105	−32.951	0.000	−32.951	−0.08514	0.00000	−0.08514	−214.43	−0.4663	1.383	−137
−136	3.229	8.130	0.000	8.130	−32.709	0.000	−32.709	−0.08440	0.00000	−0.08440	−214.09	−0.4653	1.537	−136
−135	3.586	8.156	0.000	8.156	−32.468	0.000	−32.468	−0.08365	0.00000	−0.08365	−213.75	−0.4642	1.707	−135
−134	3.980	8.181	0.000	8.181	−32.226	0.000	−32.226	−0.08291	0.00000	−0.08291	−213.41	−0.4632	1.895	−134
−133	4.414	8.207	0.000	8.207	−31.985	0.000	−31.985	−0.08217	0.00000	−0.08217	−213.07	−0.4621	2.102	−133
−132	4.893	8.232	0.000	8.232	−31.744	0.000	−31.744	−0.08144	0.00000	−0.08144	−212.72	−0.4611	2.330	−132
−131	5.419	8.258	0.000	8.258	−31.503	0.000	−31.503	−0.08070	0.00000	−0.08070	−212.38	−0.4600	2.581	−131
−130	6.000	8.283	0.000	8.283	−31.262	0.000	−31.262	−0.07997	0.00000	−0.07997	−212.03	−0.4590	2.858	−130
−129	6.637	8.309	0.000	8.309	−31.021	0.000	−31.021	−0.07924	0.00000	−0.07924	−211.68	−0.4579	3.182	−129

[a] Compiled by John A. Goff and S. Gratch.

TABLE 4—THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE 29.921 IN. HG)—*Continued*

FAHR. TEMP. t(F)	HUMIDITY RATIO $W_s \times 10^7$	VOLUME CU FT/LB DRY AIR			ENTHALPY BTU/LB DRY AIR			ENTROPY BTU PER (°F) (LB DRY AIR)			CONDENSED WATER			FAHR. TEMP. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h'_w	Entropy Btu/(Lb)(°F) s'_w	Vap. Press. In. Hg $p_s \times 10^5$	
−128	0.7339	8.334	0.000	8.334	−30.780	0.000	−30.780	−0.07851	0.00000	−0.07851	−211.33	−0.4569	0.3492	−128
−127	0.8111	8.360	0.000	8.360	−30.539	0.000	−30.539	−0.07778	0.00000	−0.07778	−210.98	−0.4559	0.3863	−127
−126	0.8958	8.385	0.000	8.385	−30.298	0.000	−30.298	−0.07707	0.00000	−0.07707	−210.63	−0.4548	0.4267	−126
−125	0.9887	8.411	0.000	8.411	−30.057	0.000	−30.057	−0.07634	0.00000	−0.07634	−210.28	−0.4538	0.4710	−125
−124	1.091	8.436	0.000	8.436	−29.816	0.000	−29.816	−0.07562	0.00000	−0.07562	−209.93	−0.4527	0.5197	−124
−123	1.202	8.461	0.000	8.461	−29.575	0.000	−29.575	−0.07490	0.00000	−0.07490	−209.58	−0.4517	0.5730	−123
−122	1.325	8.486	0.000	8.486	−29.334	0.000	−29.334	−0.07419	0.00000	−0.07419	−209.23	−0.4506	0.6314	−122
−121	1.459	8.512	0.000	8.512	−29.093	0.000	−29.093	−0.07348	0.00000	−0.07348	−208.88	−0.4496	0.6953	−121
−120	1.606	8.537	0.000	8.537	−28.852	0.000	−28.852	−0.07277	0.00000	−0.07277	−208.52	−0.4485	0.7653	−120
−119	1.767	8.563	0.000	8.563	−28.611	0.000	−28.611	−0.07206	0.00000	−0.07206	−208.17	−0.4475	0.8419	−119
−118	1.942	8.588	0.000	8.588	−28.370	0.000	−28.370	−0.07135	0.00000	−0.07135	−207.81	−0.4464	0.9256	−118
−117	2.134	8.613	0.000	8.613	−28.129	0.000	−28.129	−0.07064	0.00000	−0.07064	−207.45	−0.4454	1.017	−117
−116	2.343	8.639	0.000	8.639	−27.889	0.000	−27.889	−0.06994	0.00000	−0.06994	−207.09	−0.4444	1.117	−116
−115	2.571	8.664	0.000	8.664	−27.648	0.000	−27.648	−0.06924	0.00000	−0.06924	−206.73	−0.4433	1.226	−115
−114	2.820	8.690	0.000	8.690	−27.407	0.000	−27.407	−0.06854	0.00000	−0.06854	−206.37	−0.4423	1.345	−114
−113	3.092	8.715	0.000	8.715	−27.166	0.000	−27.166	−0.06784	0.00000	−0.06784	−206.01	−0.4412	1.475	−113
−112	3.388	8.741	0.000	8.741	−26.926	0.000	−26.926	−0.06715	0.00000	−0.06715	−205.65	−0.4402	1.617	−112
−111	3.711	8.766	0.000	8.766	−26.685	0.000	−26.685	−0.06646	0.00000	−0.06646	−205.29	−0.4392	1.771	−111
−110	4.063	8.792	0.000	8.792	−26.444	0.000	−26.444	−0.06577	0.00000	−0.06577	−204.92	−0.4381	1.939	−110
−109	4.445	8.817	0.000	8.817	−26.204	0.000	−26.204	−0.06508	0.00000	−0.06508	−204.46	−0.4371	2.121	−109
−108	4.861	8.842	0.000	8.842	−25.963	0.001	−25.962	−0.06439	0.00000	−0.06439	−204.19	−0.4360	2.320	−108
−107	5.314	8.868	0.000	8.868	−25.722	0.001	−25.721	−0.06370	0.00000	−0.06370	−203.83	−0.4350	2.536	−107
−106	5.806	8.893	0.000	8.893	−25.481	0.001	−25.480	−0.06302	0.00000	−0.06302	−203.46	−0.4339	2.771	−106
−105	6.340	8.919	0.000	8.919	−25.240	0.001	−25.239	−0.06234	0.00000	−0.06234	−203.09	−0.4329	3.026	−105
−104	6.920	8.944	0.000	8.944	−25.000	0.001	−24.999	−0.06167	0.00000	−0.06167	−202.72	−0.4318	3.303	−104
−103	7.549	8.970	0.000	8.970	−24.759	0.001	−24.758	−0.06099	0.00000	−0.06099	−202.35	−0.4308	3.603	−103
−102	8.232	8.995	0.000	8.995	−24.518	0.001	−24.517	−0.06032	0.00000	−0.06032	−201.98	−0.4298	3.929	−102
−101	8.972	9.020	0.000	9.020	−24.278	0.001	−24.277	−0.05964	0.00000	−0.05964	−201.61	−0.4287	4.283	−101
−100	9.772	9.046	0.000	9.046	−24.037	0.001	−24.036	−0.05897	0.00000	−0.05897	−201.23	−0.4277	4.666	−100
−99	10.63	9.071	0.000	9.071	−23.797	0.001	−23.796	−0.05830	0.00000	−0.05830	−200.86	−0.4266	5.081	−99
−98	11.57	9.097	0.000	9.097	−23.556	0.001	−23.555	−0.05764	0.00000	−0.05764	−200.48	−0.4256	5.530	−98
−97	12.59	9.122	0.000	9.122	−23.316	0.001	−23.315	−0.05697	0.00000	−0.05697	−200.11	−0.4245	6.016	−97

[a] Compiled by John A. Goff and S. Gratch.

TABLE 4—THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE 29.921 IN. HG)—*Continued*

FAHR. TEMP. t(F)	HUMIDITY RATIO $W_s \times 10^6$	VOLUME CU FT/LB DRY AIR v_a	v_{as}	v_s	ENTHALPY BTU/LB DRY AIR h_a	h_{as}	h_s	ENTROPY BTU PER (°F) (LB DRY AIR) s_a	s_{as}	s_s	CONDENSED WATER Enthalpy Btu/Lb h'_w	Entropy Btu/(Lb)(°F) s'_w	Vap. Press. In. Hg $p_s \times 10^4$	FAHR. TEMP. t(F)
−96	1.369	9.147	0.000	9.147	−23.075	0.001	−23.074	−0.05631	0.00000	−0.05631	−199.73	−0.4235	0.6542	−96
−95	1.489	9.173	0.000	9.173	−22.835	0.002	−22.833	−0.05565	0.00000	−0.05565	−199.35	−0.4225	0.7111	−95
−94	1.617	9.198	0.000	9.198	−22.594	0.002	−22.592	−0.05500	0.00000	−0.05500	−198.97	−0.4214	0.7725	−94
−93	1.756	9.224	0.000	9.224	−22.353	0.002	−22.351	−0.05434	0.00000	−0.05434	−198.59	−0.4204	0.8388	−93
−92	1.906	9.249	0.000	9.249	−22.113	0.002	−22.111	−0.05369	0.00001	−0.05368	−198.21	−0.4193	0.9105	−92
−91	2.068	9.275	0.000	9.275	−21.872	0.002	−21.870	−0.05303	0.00001	−0.05302	−197.83	−0.4183	0.9879	−91
−90	2.242	9.300	0.000	9.300	−21.631	0.002	−21.629	−0.05237	0.00001	−0.05236	−197.44	−0.4173	1.071	−90
−89	2.430	9.325	0.000	9.325	−21.391	0.003	−21.388	−0.05172	0.00001	−0.05171	−197.06	−0.4162	1.161	−89
−88	2.634	9.351	0.000	9.351	−21.150	0.003	−21.147	−0.05107	0.00001	−0.05106	−196.67	−0.4152	1.259	−88
−87	2.852	9.376	0.000	9.376	−20.909	0.003	−20.906	−0.05042	0.00001	−0.05041	−196.29	−0.4142	1.363	−87
−86	3.089	9.401	0.000	9.401	−20.669	0.003	−20.666	−0.04978	0.00001	−0.04977	−195.90	−0.4131	1.476	−86
−85	3.342	9.426	0.000	9.426	−20.428	0.003	−20.425	−0.04913	0.00001	−0.04912	−195.51	−0.4121	1.597	−85
−84	3.615	9.451	0.000	9.451	−20.188	0.004	−20.184	−0.04849	0.00001	−0.04848	−195.12	−0.4110	1.728	−84
−83	3.909	9.477	0.000	9.477	−19.947	0.004	−19.943	−0.04785	0.00001	−0.04784	−194.73	−0.4100	1.868	−83
−82	4.225	9.502	0.000	9.502	−19.706	0.004	−19.702	−0.04721	0.00001	−0.04720	−194.34	−0.4090	2.019	−82
−81	4.564	9.527	0.000	9.527	−19.466	0.005	−19.461	−0.04658	0.00001	−0.04657	−193.95	−0.4079	2.181	−81
−80	4.930	9.553	0.000	9.553	−19.225	0.005	−19.220	−0.04595	0.00001	−0.04594	−193.55	−0.4069	2.356	−80
−79	5,322	9.578	0.000	9.578	−18.984	0.005	−18.979	−0.04532	0.00002	−0.04530	−193.16	−0.4059	2.543	−79
−78	5.742	9.604	0.000	9.604	−18.744	0.006	−18.738	−0.04469	0.00002	−0.04467	−192.76	−0.4048	2.744	−78
−77	6.193	9.629	0.000	9.629	−18.503	0.006	−18.497	−0.04406	0.00002	−0.04404	−192.37	−0.4038	2.960	−77
−76	6.677	9.654	0.000	9.654	−18.263	0.007	−18.256	−0.04343	0.00002	−0.04341	−191.97	−0.4027	3.192	−76
−75	7.196	9.680	0.000	9.680	−18.022	0.007	−18.015	−0.04280	0.00002	−0.04278	−191.57	−0.4017	3.441	−75
−74	7.753	9.705	0.000	9.705	−17.782	0.008	−17.774	−0.04218	0.00003	−0.04215	−191.17	−0.4007	3.707	−74
−73	8.349	9.730	0.000	9.730	−17.541	0.008	−17.533	−0.04155	0.00003	−0.04152	−190.77	−0.3996	3.992	−73
−72	8.990	9.756	0.000	9.756	−17.301	0.009	−17.292	−0.04093	0.00003	−0.04090	−190.37	−0.3986	4.298	−72
−71	9.675	9.781	0.000	9.781	−17.060	0.010	−17.050	−0.04031	0.00003	−0.04028	−189.97	−0.3975	4.625	−71
−70	10.40	9.806	0.000	9.806	−16.820	0.011	−16.809	−0.03969	0.00003	−0.03966	−189.56	−0.3965	4.976	−70
−69	11.19	9.831	0.000	9.831	−16.579	0.011	−16.568	−0.03908	0.00004	−0.03904	−189.16	−0.3954	5.351	−69
−68	12.03	9.856	0.000	9.856	−16.339	0.012	−16.327	−0.03846	0.00004	−0.03842	−188.75	−0.3944	5.752	−68
−67	12.92	9.882	0.000	9.882	−16.098	0.013	−16.085	−0.03785	0.00004	−0.03781	−188.35	−0.3934	6.181	−67
−66	13.88	9.907	0.000	9.907	−15.858	0.014	−15.844	−0.03724	0.00004	−0.03720	−187.94	−0.3924	6.640	−66
−65	14.91	9.932	0.000	9.932	−15.617	0.015	−15.602	−0.03663	0.00005	−0.03658	−187.53	−0.3913	7.130	−65

[a] Compiled by John A. Goff and S. Gratch.

TABLE 4—THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE 29.921 IN. HG)—*Continued*

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^5$	Volume cu ft/lb dry air v_a	v_{as}	v_s	Enthalpy Btu/lb dry air h_a	h_{as}	h_s	Entropy Btu per (°F) (lb dry air) s_a	s_{as}	s_s	Condensed Water Enthalpy Btu/Lb h'_w	Entropy Btu/(Lb)(°F) s'_w	Vap. Press. In. Hg $p_s \times 10^3$	Fahr. Temp. t(F)
−64	1.601	9.958	0.000	9.958	−15.377	0.016	−15.361	−0.03602	0.00005	−0.03597	−187.12	−0.3903	0.7654	−64
−63	1.718	9.983	0.000	9.983	−15.137	0.018	−15.119	−0.03541	0.00005	−0.03536	−186.71	−0.3893	0.8213	−63
−62	1.843	10.009	0.000	10.009	−14.896	0.019	−14.877	−0.03481	0.00006	−0.03475	−186.30	−0.3882	0.8810	−62
−61	1.976	10.034	0.000	10.034	−14.656	0.020	−14.636	−0.03420	0.00006	−0.03414	−185.89	−0.3872	0.9447	−61
−60	2.118	10.059	0.000	10.059	−14.416	0.022	−14.394	−0.03360	0.00006	−0.03354	−185.47	−0.3861	1.0127	−60
−59	2.269	10.085	0.000	10.085	−14.175	0.023	−14.152	−0.03300	0.00007	−0.03293	−185.06	−0.3851	1.0852	−59
−58	2.431	10.110	0.000	10.110	−13.935	0.025	−13.910	−0.03240	0.00007	−0.03233	−184.64	−0.3841	1.1624	−58
−57	2.603	10.135	0.000	10.135	−13.695	0.027	−13.668	−0.03180	0.00008	−0.03172	−184.23	−0.3830	1.2447	−57
−56	2.786	10.161	0.000	10.161	−13.454	0.029	−13.425	−0.03120	0.00008	−0.03112	−183.81	−0.3820	1.3324	−56
−55	2.982	10.186	0.000	10.186	−13.214	0.031	−13.183	−0.03061	0.00009	−0.03052	−183.39	−0.3810	1.4258	−55
−54	3.190	10.211	0.000	10.211	−12.974	0.033	−12.941	−0.03002	0.00009	−0.02993	−182.97	−0.3799	1.5253	−54
−53	3.411	10.237	0.000	10.237	−12.733	0.035	−12.698	−0.02943	0.00010	−0.02933	−182.55	−0.3789	1.6312	−53
−52	3.646	10.262	0.001	10.263	−12.493	0.038	−12.455	−0.02884	0.00011	−0.02873	−182.13	−0.3779	1.7438	−52
−51	3.897	10.288	0.001	10.289	−12.253	0.041	−12.212	−0.02825	0.00011	−0.02814	−181.71	−0.3769	1.8635	−51
−50	4.163	10.313	0.001	10.314	−12.012	0.043	−11.969	−0.02766	0.00012	−0.02754	−181.29	−0.3758	1.9910	−50
−49	4.446	10.338	0.001	10.339	−11.772	0.046	−11.726	−0.02707	0.00012	−0.02695	−180.87	−0.3748	2.1264	−49
−48	4.747	10.364	0.001	10.365	−11.532	0.049	−11.483	−0.02649	0.00013	−0.02636	−180.44	−0.3738	2.2702	−48
−47	5.066	10.389	0.001	10.390	−11.292	0.053	−11.239	−0.02590	0.00013	−0.02577	−180.02	−0.3728	2.4230	−47
−46	5.406	10.414	0.001	10.415	−11.051	0.056	−10.995	−0.02532	0.00014	−0.02518	−179.59	−0.3717	2.5854	−46
−45	5.766	10.440	0.001	10.441	−10.811	0.060	−10.751	−0.02474	0.00015	−0.02459	−179.16	−0.3707	2.7578	−45
−44	6.149	10.465	0.001	10.466	−10.571	0.064	−10.507	−0.02416	0.00016	−0.02400	−178.73	−0.3696	2.9408	−44
−43	6.555	10.490	0.001	10.491	−10.330	0.068	−10.262	−0.02358	0.00017	−0.02341	−178.30	−0.3686	3.1349	−43
−42	6.985	10.516	0.001	10.517	−10.090	0.073	−10.017	−0.02301	0.00019	−0.02282	−177.87	−0.3676	3.3408	−42
−41	7.441	10.541	0.001	10.542	−9.850	0.078	−9.772	−0.02243	0.00020	−0.02223	−177.44	−0.3665	3.5591	−41
−40	7.925	10.566	0.001	10.567	−9.609	0.083	−9.526	−0.02186	0.00021	−0.02165	−177.01	−0.3655	3.7906	−40
−39	8.437	10.592	0.001	10.593	−9.369	0.089	−9.280	−0.02129	0.00022	−0.02107	−176.58	−0.3645	4.0359	−39
−38	8.980	10.617	0.002	10.619	−9.129	0.094	−9.035	−0.02072	0.00024	−0.02048	−176.14	−0.3634	4.2958	−38
−37	9.556	10.642	0.002	10.644	−8.889	0.100	−8.789	−0.02015	0.00025	−0.01990	−175.71	−0.3624	4.5711	−37
−36	10.16	10.668	0.002	10.670	−8.648	0.106	−8.542	−0.01958	0.00026	−0.01932	−175.27	−0.3614	4.8626	−36
−35	10.81	10.693	0.002	10.695	−8.408	0.113	−8.295	−0.01902	0.00028	−0.01874	−174.84	−0.3604	5.1713	−35
−34	11.49	10.718	0.002	10.720	−8.168	0.121	−8.047	−0.01845	0.00030	−0.01815	−174.40	−0.3593	5.4980	−34
−33	12.21	10.744	0.002	10.746	−7.927	0.128	−7.799	−0.01789	0.00032	−0.01757	−173.96	−0.3583	5.8437	−33

[a] Compiled by John A. Goff and S. Gratch.

TABLE 4—THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG)—*Continued*

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^4$	Volume cu ft/lb dry air v_a	v_{as}	v_s	Enthalpy Btu/lb dry air h_a	h_{as}	h_s	Entropy Btu per (°F) (lb dry air) s_a	s_{as}	s_s	Condensed Water Enthalpy Btu/Lb h'_w	Entropy Btu/(Lb)(°F) s'_w	Vap. Press. In. Hg $p_s \times 10^3$	Fahr. Temp. t(F)
−32	1.298	10.769	0.002	10.771	−7.687	0.136	−7.551	−0.01733	0.00034	−0.01699	−173.52	−0.3573	0.62093	−32
−31	1.378	10.794	0.002	10.796	−7.447	0.145	−7.302	−0.01677	0.00036	−0.01641	−173.08	−0.3563	0.65979	−31
−30	1.464	10.820	0.002	10.822	−7.207	0.154	−7.053	−0.01621	0.00038	−0.01583	−172.64	−0.3552	0.70046	−30
−29	1.554	10.845	0.003	10.848	−6.966	0.163	−6.803	−0.01565	0.00040	−0.01525	−172.20	−0.3542	0.74365	−29
−28	1.649	10.870	0.003	10.873	−6.726	0.173	−6.553	−0.01509	0.00043	−0.01466	−171.75	−0.3531	0.78928	−28
−27	1.750	10.896	0.003	10.899	−6.486	0.184	−6.302	−0.01453	0.00045	−0.01408	−171.31	−0.3521	0.83748	−27
−26	1.856	10.921	0.003	10.924	−6.246	0.196	−6.050	−0.01398	0.00048	−0.01350	−170.86	−0.3511	0.88838	−26
−25	1.969	10.946	0.004	10.950	−6.005	0.207	−5.798	−0.01342	0.00051	−0.01291	−170.42	−0.3500	0.94212	−25
−24	2.087	10.972	0.004	10.976	−5.765	0.219	−5.546	−0.01287	0.00054	−0.01233	−169.97	−0.3490	0.99885	−24
−23	2.212	10.997	0.004	11.001	−5.525	0.232	−5.293	−0.01232	0.00057	−0.01175	−169.52	−0.3480	1.0587	−23
−22	2.344	11.022	0.004	11.026	−5.285	0.246	−5.039	−0.01177	0.00061	−0.01116	−169.07	−0.3469	1.1219	−22
−21	2.483	11.048	0.004	11.052	−5.044	0.261	−4.783	−0.01122	0.00064	−0.01058	−168.62	−0.3459	1.1885	−21
−20	2.630	11.073	0.005	11.078	−4.804	0.277	−4.527	−0.01067	0.00068	−0.00999	−168.17	−0.3449	1.2587	−20
−19	2.785	11.098	0.005	11.103	−4.564	0.293	−4.271	−0.01012	0.00072	−0.00940	−167.72	−0.3439	1.3327	−19
−18	2.948	11.124	0.005	11.129	−4.324	0.310	−4.014	−0.00958	0.00076	−0.00882	−167.26	−0.3428	1.4107	−18
−17	3.120	11.149	0.006	11.155	−4.083	0.328	−3.755	−0.00904	0.00080	−0.00824	−166.81	−0.3418	1.4929	−17
−16	3.301	11.174	0.006	11.180	−3.843	0.348	−3.495	−0.00850	0.00084	−0.00766	−166.35	−0.3408	1.5795	−16
−15	3.491	11.200	0.006	11.206	−3.603	0.368	−3.235	−0.00796	0.00089	−0.00707	−165.90	−0.3398	1.6706	−15
−14	3.692	11.225	0.007	11.232	−3.363	0.389	−2.974	−0.00743	0.00094	−0.00649	−165.44	−0.3387	1.7666	−14
−13	3.903	11.250	0.007	11.257	−3.123	0.412	−2.711	−0.00689	0.00099	−0.00590	−164.98	−0.3377	1.8677	−13
−12	4.125	11.275	0.008	11.283	−2.882	0.436	−2.446	−0.00636	0.00104	−0.00532	−164.52	−0.3367	1.9740	−12
−11	4.359	11.301	0.008	11.309	−2.642	0.461	−2.181	−0.00582	0.00109	−0.00473	−164.06	−0.3357	2.0859	−11
−10	4.606	11.326	0.008	11.334	−2.402	0.487	−1.915	−0.00529	0.00115	−0.00414	−163.60	−0.3346	2.2035	−10
−9	4.865	11.351	0.008	11.359	−2.162	0.514	−1.648	−0.00475	0.00121	−0.00354	−163.14	−0.3336	2.3272	−9
−8	5.137	11.376	0.009	11.385	−1.922	0.543	−1.379	−0.00422	0.00128	−0.00294	−162.67	−0.3326	2.4573	−8
−7	5.423	11.401	0.010	11.411	−1.681	0.574	−1.107	−0.00369	0.00135	−0.00234	−162.21	−0.3316	2.5940	−7
−6	5.724	11.427	0.010	11.437	−1.441	0.606	−0.835	−0.00316	0.00142	−0.00174	−161.74	−0.3305	2.7377	−6
−5	6.040	11.452	0.011	11.463	−1.201	0.639	−0.562	−0.00263	0.00149	−0.00114	−161.28	−0.3295	2.8886	−5
−4	6.371	11.477	0.012	11.489	−0.961	0.675	−0.286	−0.00210	0.00157	−0.00053	−160.81	−0.3285	3.0472	−4
−3	6.720	11.502	0.013	11.515	−0.721	0.712	−0.009	−0.00157	0.00165	0.00008	−160.34	−0.3275	3.2137	−3
−2	7.085	11.528	0.013	11.541	−0.480	0.751	0.271	−0.00105	0.00174	0.00069	−159.87	−0.3264	3.3885	−2
−1	7.469	11.553	0.014	11.567	−0.240	0.792	0.552	−0.00052	0.00183	0.00131	−159.40	−0.3254	3.5720	−1

[a] Compiled by John A. Goff and S. Gratch.

TABLE 4—THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG)—*Continued*

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^3$	Volume cu ft/lb dry air v_a	v_{as}	v_s	Enthalpy Btu/lb dry air h_a	h_{as}	h_s	Entropy Btu per (°F) (lb dry air) s_a	s_{as}	s_s	Condensed Water Enthalpy Btu/Lb h'_w	Entropy Btu/(LB)(°F) s'_w	Vap. Press. In. Hg $p_s \times 10^3$	Fahr. Temp. t(F)
0	0.7872	11.578	0.015	11.593	0.000	0.835	0.835	0.00000	0.00192	0.00192	−158.93	−0.3244	3.7645	0
1	0.8295	11.604	0.015	11.619	0.240	0.880	1.120	0.00052	0.00202	0.00254	−158.46	−0.3234	3.9666	1
2	0.8739	11.629	0.016	11.645	0.480	0.928	1.408	0.00104	0.00212	0.00316	−157.99	−0.3223	4.1785	2
3	0.9204	11.654	0.017	11.671	0.721	0.977	1.698	0.00156	0.00223	0.00379	−157.52	−0.3213	4.4007	3
4	0.9692	11.679	0.018	11.697	0.961	1.030	1.991	0.00208	0.00234	0.00442	−157.04	−0.3203	4.6337	4
5	1.020	11.705	0.019	11.724	1.201	1.085	2.286	0.00260	0.00246	0.00506	−156.57	−0.3193	4.8779	5
6	1.074	11.730	0.020	11.750	1.441	1.142	2.583	0.00312	0.00258	0.00570	−156.09	−0.3182	5.1339	6
7	1.130	11.756	0.021	11.777	1.681	1.202	2.883	0.00364	0.00271	0.00635	−155.61	−0.3172	5.4022	7
8	1.189	11.781	0.022	11.803	1.922	1.266	3.188	0.00415	0.00285	0.00700	−155.13	−0.3162	5.6832	8
9	1.251	11.806	0.024	11.830	2.162	1.332	3.494	0.00467	0.00299	0.00766	−154.65	−0.3152	5.9776	9
10	1.315	11.831	0.025	11.856	2.402	1.401	3.803	0.00518	0.00314	0.00832	−154.17	−0.3141	6.2858	10
11	1.383	11.857	0.026	11.883	2.642	1.474	4.116	0.00569	0.00330	0.00899	−153.69	−0.3131	6.6085	11
12	1.454	11.882	0.028	11.910	2.882	1.550	4.432	0.00620	0.00346	0.00966	−153.21	−0.3121	6.9462	12
13	1.528	11.907	0.029	11.936	3.123	1.630	4.753	0.00671	0.00363	0.01034	−152.73	−0.3111	7.2997	13
14	1.606	11.933	0.030	11.963	3.363	1.713	5.076	0.00721	0.00380	0.01101	−152.24	−0.3100	7.6696	14
15	1.687	11.958	0.032	11.990	3.603	1.800	5.403	0.00772	0.00399	0.01171	−151.76	−0.3090	8.0565	15
16	1.772	11.983	0.034	12.017	3.843	1.892	5.735	0.00822	0.00418	0.01240	−151.27	−0.3080	8.4612	16
17	1.861	12.009	0.035	12.044	4.083	1.988	6.071	0.00873	0.00438	0.01311	−150.78	−0.3070	8.8843	17
18	1.953	12.034	0.038	12.072	4.324	2.088	6.412	0.00923	0.00459	0.01382	−150.29	−0.3059	9.3267	18
19	2.051	12.059	0.040	12.099	4.564	2.192	6.756	0.00973	0.00481	0.01454	−149.80	−0.3049	9.7889	19
20	2.152	12,084	0.042	12,126	4.804	2.302	7.106	0.01023	0.00504	0.01527	−149.31	−0.3039	10.272	20
21	2.258	12.110	0.044	12.154	5.044	2.416	7.460	0.01073	0.00528	0.01601	−148.82	−0.3029	10.777	21
22	2.369	12.135	0.046	12.181	5.284	2.536	7.820	0.01123	0.00553	0.01676	−148.33	−0.3018	11.305	22
23	2.485	12.160	0.049	12.209	5.525	2.661	8.186	0.01173	0.00579	0.01752	−147.84	−0.3008	11.856	23
24	2.606	12.186	0.051	12.237	5.765	2.792	8.557	0.01223	0.00607	0.01830	−147.34	−0.2998	12.431	24
25	2.733	12.211	0.054	12.265	6.005	2.929	8.934	0.01273	0.00635	0.01908	−146.85	−0.2988	13.032	25
26	2.865	12.236	0.057	12.293	6.245	3.072	9.317	0.01322	0.00665	0.01987	−146.35	−0.2977	13.659	26
27	3.003	12.262	0.059	12.321	6.485	3.221	9.706	0.01372	0.00696	0.02068	−145.85	−0.2967	14.313	27
28	3.147	12.287	0.062	12.349	6.726	3.377	10.103	0.01421	0.00728	0.02149	−145.36	−0.2957	14.996	28
29	3.297	12.312	0.065	12.377	6.966	3.540	10.506	0.01470	0.00761	0.02231	−144.86	−0.2947	15.709	29
30	3,454	12.338	0.068	12.406	7.206	3.709	10.915	0.01519	0.00796	0.02315	−144.36	−0.2936	16.452	30
31	3.617	12.363	0.071	12.434	7.446	3.887	11.333	0.01568	0.00832	0.02400	−143.86	−0.2926	17.227	31
32	3.788	12.388	0.075	12.463	7.686	4.072	11.758	0.01617	0.00870	0.02487	−143.36	−0.2916	18.035	32
32[b]	3.788	12.388	0.075	12.463	7.686	4.072	11.758	0.01617	0.00870	0.02487	0.04	0.0000	18.037	32[b]
33	3.944	12.413	0.079	12.492	7.927	4.242	12.169	0.01666	0.00904	0.02570	1.05	0.0020	18.778	33
34	4.107	12.438	0.082	12.520	8.167	4.418	12.585	0.01715	0.00940	0.02655	2.06	0.0041	19.546	34

[a] Compiled by John A. Goff and S. Gratch.
[b] Extrapolated to represent metastable equilibrium with undercooled liquid.

TABLE 4—THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG)—*Continued*

FAHR. TEMP. t(F)	HUMIDITY RATIO $W_s \times 10^3$	VOLUME CU FT/LB DRY AIR			ENTHALPY BTU/LB DRY AIR			ENTROPY BTU PER (°F) (LB DRY AIR)			CONDENSED WATER			FAHR. TEMP. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h'_w	Entropy Btu/(Lb)(°F) s'_w	Vap. Press. In. Hg p_s	
35	4.275	12.464	0.085	12.549	8.407	4.601	13.008	0.01764	0.00977	0.02741	3.06	0.0061	0.20342	35
36	4.450	12.489	0.089	12.578	8.647	4.791	13.438	0.01812	0.01016	0.02828	4.07	0.0081	0.21166	36
37	4.631	12.514	0.093	12.607	8.887	4.987	13.874	0.01861	0.01056	0.02917	5.07	0.0102	0.22020	37
38	4.818	12.540	0.097	12.637	9.128	5.191	14.319	0.01909	0.01097	0.03006	6.08	0.0122	0.22904	38
39	5.012	12.565	0.101	12.666	9.368	5.403	14.771	0.01957	0.01139	0.03096	7.08	0.0142	0.23819	39
40	5.213	12.590	0.105	12.695	9.608	5.622	15.230	0.02005	0.01183	0.03188	8.09	0.0162	0.24767	40
41	5.421	12.616	0.109	12.725	9.848	5.849	15.697	0.02053	0.01228	0.03281	9.09	0.0182	0.25748	41
42	5.638	12.641	0.114	12.755	10.088	6.084	16.172	0.02101	0.01275	0.03376	10.09	0.0202	0.26763	42
43	5.860	12.666	0.119	12.785	10.329	6.328	16.657	0.02149	0.01323	0.03472	11.10	0.0222	0.27813	43
44	6.091	12.691	0.124	12.815	10.569	6.580	17.149	0.02197	0.01373	0.03570	12.10	0.0242	0.28899	44
45	6.331	12.717	0.129	12.846	10.809	6.841	17.650	0.02245	0.01425	0.03670	13.10	0.0262	0.30023	45
46	6.578	12.742	0.134	12.876	11.049	7.112	18.161	0.02293	0.01478	0.03771	14.10	0.0282	0.31185	46
47	6.835	12.767	0.140	12.907	11.289	7.391	18.680	0.02340	0.01534	0.03874	15.11	0.0302	0.32386	47
48	7.100	12.792	0.146	12.938	11.530	7.681	19.211	0.02387	0.01591	0.03978	16.11	0.0321	0.33629	48
49	7.374	12.818	0.151	12.969	11.770	7.981	19.751	0.02434	0.01650	0.04084	17.11	0.0341	0.34913	49
50	7.658	12.843	0.158	13.001	12.010	8.291	20.301	0.02481	0.01711	0.04192	18.11	0.0361	0.36240	50
51	7.952	12.868	0.164	13.032	12.250	8.612	20.862	0.02528	0.01774	0.04302	19.11	0.0381	0.37611	51
52	8.256	12.894	0.170	13.064	12.491	8.945	21.436	0.02575	0.01839	0.04414	20.11	0.0400	0.39028	52
53	8.569	12.919	0.178	13.097	12.731	9.289	22.020	0.02622	0.01906	0.04528	21.12	0.0420	0.40492	53
54	8.894	12.944	0.185	13.129	12.971	9.644	22.615	0.02669	0.01976	0.04645	22.12	0.0439	0.42004	54
55	9.229	12.970	0.192	13.162	13.211	10.01	23.22	0.02716	0.02047	0.04763	23.12	0.0459	0.43565	55
56	9.575	12.995	0.200	13.195	13.452	10.39	23.84	0.02762	0.02121	0.04883	24.12	0.0478	0.45176	56
57	9.934	13.020	0.208	13.228	13.692	10.79	24.48	0.02809	0.02197	0.05006	25.12	0.0497	0.46840	57
58	10.30	13.045	0.216	13.261	13.932	11.19	25.12	0.02855	0.02276	0.05131	26.12	0.0517	0.48558	58
59	10.69	13.071	0.224	13.295	14.172	11.61	25.78	0.02902	0.02357	0.05259	27.12	0.0536	0.50330	59
60	11.08	13.096	0.233	13.329	14.413	12.05	26.46	0.02948	0.02441	0.05389	28.12	0.0555	0.52159	60
61	11.49	13.121	0.242	13.363	14.653	12.50	27.15	0.02994	0.02527	0.05521	29.12	0.0574	0.54047	61
62	11.91	13.147	0.251	13.398	14.893	12.96	27.85	0.03040	0.02616	0.05656	30.12	0.0594	0.55994	62
63	12.35	13.172	0.261	13.433	15.134	13.44	28.57	0.03086	0.02708	0.05794	31.12	0.0613	0.58002	63
64	12.80	13.197	0.271	13.468	15.374	13.94	29.31	0.03132	0.02803	0.05935	32.12	0.0632	0.60073	64
65	13.26	13.222	0.282	13.504	15.614	14.45	30.06	0.03177	0.02901	0.06078	33.11	0.0651	0.62209	65
66	13.74	13.247	0.292	13.539	15.855	14.98	30.83	0.03223	0.03002	0.06225	34.11	0.0670	0.64411	66
67	14.24	13.273	0.303	13.576	16.095	15.53	31.62	0.03269	0.03106	0.06375	35.11	0.0689	0.66681	67
68	14.75	13.298	0.315	13.613	16.335	16.09	32.42	0.03314	0.03213	0.06527	36.11	0.0708	0.69019	68
69	15.28	13.323	0.327	13.650	16.576	16.67	33.25	0.03360	0.03323	0.06683	37.11	0.0727	0.71430	69

[a] Compiled by John A. Goff and S. Gratch.

TABLE 4—THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG)—*Continued*

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^2$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h'_w	Entropy Btu/(Lb)(°F) s'_w	Vap. Press. In. Hg p_s	
70	1.582	13.348	0.339	13.687	16.816	17.27	34.09	0.03405	0.03437	0.06842	38.11	0.0746	0.73915	70
71	1.639	13.373	0.351	13.724	17.056	17.89	34.95	0.03450	0.03554	0.07004	39.11	0.0765	0.76475	71
72	1.697	13.398	0.364	13.762	17.297	18.53	35.83	0.03495	0.03675	0.07170	40.11	0.0784	0.79112	72
73	1.757	13.424	0.377	13.801	17.537	19.20	36.74	0.03540	0.03800	0.07340	41.11	0.0803	0.81828	73
74	1.819	13.449	0.392	13.841	17.778	19.88	37.66	0.03585	0.03928	0.07513	42.10	0.0821	0.84624	74
75	1.882	13.474	0.407	13.881	18.018	20.59	38.61	0.03630	0.04060	0.07690	43.10	0.0840	0.87504	75
76	1.948	13.499	0.422	13.921	18.259	21.31	39.57	0.03675	0.04197	0.07872	44.10	0.0859	0.90470	76
77	2.016	13.525	0.437	13.962	18.499	22.07	40.57	0.03720	0.04337	0.08057	45.10	0.0877	0.93523	77
78	2.086	13.550	0.453	14.003	18.740	22.84	41.58	0.03765	0.04482	0.08247	46.10	0.0896	0.96665	78
79	2.158	13.575	0.470	14.045	18.980	23.64	42.62	0.03810	0.04631	0.08441	47.10	0.0914	0.99899	79
80	2.233	13.601	0.486	14.087	19.221	24.47	43.69	0.03854	0.04784	0.08638	48.10	0.0933	1.0323	80
81	2.310	13.626	0.504	14.130	19.461	25.32	44.78	0.03899	0.04942	0.08841	49.09	0.0952	1.0665	81
82	2.389	13.651	0.523	14.174	19.702	26.20	45.90	0.03943	0.05105	0.09048	50.09	0.0970	1.1017	82
83	2.471	13.676	0.542	14.218	19.942	27.10	47.04	0.03987	0.05273	0.09260	51.09	0.0989	1.1379	83
84	2.555	13.702	0.560	14.262	20.183	28.04	48.22	0.04031	0.05446	0.09477	52.09	0.1007	1.1752	84
85	2.642	13.727	0.581	14.308	20.423	29.01	49.43	0.04075	0.05624	0.09699	53.09	0.1025	1.2135	85
86	2.731	13.752	0.602	14.354	20.663	30.00	50.66	0.04119	0.05807	0.09926	54.08	0.1043	1.2529	86
87	2.824	13.777	0.624	14.401	20.904	31.03	51.93	0.04163	0.05995	0.10158	55.08	0.1062	1.2934	87
88	2.919	13.803	0.645	14.448	21.144	32.09	53.23	0.04207	0.06189	0.10396	56.08	0.1080	1.3351	88
89	3.017	13.828	0.668	14.496	21.385	33.18	54.56	0.04251	0.06389	0.10640	57.08	0.1098	1.3779	89
90	3.118	13.853	0.692	14.545	21.625	34.31	55.93	0.04295	0.06596	0.10890	58.08	0.1116	1.4219	90
91	3.223	13.879	0.716	14.595	21.865	35.47	57.33	0.04339	0.06807	0.11146	59.07	0.1135	1.4671	91
92	3.330	13.904	0.741	14.645	22.106	36.67	58.78	0.04382	0.07025	0.11407	60.07	0.1153	1.5135	92
93	3.441	13.929	0.768	14.697	22.346	37.90	60.25	0.04426	0.07249	0.11675	61.07	0.1171	1.5612	93
94	3.556	13.954	0.795	14.749	22.587	39.18	61.77	0.04469	0.07480	0.11949	62.07	0.1188	1.6102	94
95	3.673	13.980	0.822	14.802	22.827	40.49	63.32	0.04513	0.07718	0.12231	63.07	0.1206	1.6606	95
96	3.795	14.005	0.851	14.856	23.068	41.85	64.92	0.04556	0.07963	0.12519	64.06	0.1224	1.7123	96
97	3.920	14.030	0.881	14.911	23.308	43.24	66.55	0.04600	0.08215	0.12815	65.06	0.1242	1.7654	97
98	4.049	14.056	0.911	14.967	23.548	44.68	68.23	0.04643	0.08474	0.13117	66.06	0.1260	1.8199	98
99	4.182	14.081	0.942	15.023	23.789	46.17	69.96	0.04686	0.08741	0.13427	67.06	0.1278	1.8759	99
100	4.319	14.106	0.975	15.081	24.029	47.70	71.73	0.04729	0.09016	0.13745	68.06	0.1296	1.9333	100
101	4.460	14,131	1.009	15.140	24.270	49.28	73.55	0.04772	0.09299	0.14071	69.05	0.1314	1.9923	101
102	4.606	14.157	1.043	15.200	24.510	50.91	75.42	0.04815	0.09591	0.14406	70.05	0.1332	2.0528	102
103	4.756	14.182	1.079	15.261	24.751	52.59	77.34	0.04858	0.09891	0.14749	71.05	0.1350	2.1149	103
104	4.911	14.207	1.117	15.324	24.991	54.32	79.31	0.04900	0.1020	0.1510	72.05	0.1367	2.1786	104

[a] Compiled by John A. Goff and S. Gratch.

TABLE 4—THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG)—*Continued*

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h'_w	Entropy Btu/(Lb)(°F) s'_w	Vap. Press. In. Hg p_s	
105	0.5070	14.232	1.155	15.387	25.232	56.11	81.34	0.04943	0.1052	0.1546	73.04	0.1385	2.2439	105
106	0.5234	14.258	1.194	15.452	25.472	57.95	83.42	0.04985	0.1085	0.1584	74.04	0.1403	2.3109	106
107	0.5404	14.283	1.235	15.518	25.713	59.85	85.56	0.05028	0.1118	0.1621	75.04	0.1421	2.3797	107
108	0.5578	14.308	1.278	15.586	25.953	61.80	87.76	0.05070	0.1153	0.1660	76.04	0.1438	2.4502	108
109	0.5758	14.333	1.321	15.654	26.194	63.82	90.03	0.05113	0.1189	0.1700	77.04	0.1456	2.5225	109
110	0.5944	14.359	1.365	15.724	26.434	65.91	92.34	0.05155	0.1226	0.1742	78.03	0.1472	2.5966	110
111	0.6135	14.384	1.412	15.796	26.675	68.05	94.72	0.05197	0.1264	0.1784	79.03	0.1491	2.6726	111
112	0.6333	14.409	1.460	15.869	26.915	70.27	97.18	0.05239	0.1302	0.1826	80.03	0.1508	2.7505	112
113	0.6536	14.435	1.509	15.944	27.156	72.55	99.71	0.05281	0.1342	0.1870	81.03	0.1525	2.8304	113
114	0.6746	14.460	1.560	16.020	27.397	74.91	102.31	0.05323	0.1384	0.1916	82.03	0.1543	2.9123	114
115	0.6962	14.485	1.613	16.098	27.637	77.34	104.98	0.05365	0.1426	0.1963	83.02	0.1560	2.9962	115
116	0.7185	14.510	1.668	16.178	27.878	79.85	107.73	0.05407	0.1470	0.2011	84.02	0.1577	3.0821	116
117	0.7415	14.536	1.723	16.259	28.119	82.43	110.55	0.05449	0.1515	0.2060	85.02	0.1595	3.1701	117
118	0.7652	14.561	1.782	16.343	28.359	85.10	113.46	0.05490	0.1562	0.2111	86.02	0.1612	3.2603	118
119	0.7897	14.586	1.842	16.428	28.600	87.86	116.46	0.05532	0.1610	0.2163	87.02	0.1629	3.3527	119
120	0.8149	14.611	1.905	16.516	28.841	90.70	119.54	0.05573	0.1659	0.2216	88.01	0.1646	3.4474	120
121	0.8410	14.637	1.968	16.605	29.082	93.64	122.72	0.05615	0.1710	0.2272	89.01	0.1664	3.5443	121
122	0.8678	14.662	2.034	16.696	29.322	96.66	125.98	0.05656	0.1763	0.2329	90.01	0.1681	3.6436	122
123	0.8955	14.687	2.103	16.790	29.563	99.79	129.35	0.05698	0.1817	0.2387	91.01	0.1698	3.7452	123
124	0.9242	14.712	2.174	16.886	29.804	103.0	132.8	0.05739	0.1872	0.2446	92.01	0.1715	3.8493	124
125	0.9537	14.738	2.247	16.985	30.044	106.4	136.4	0.05780	0.1930	0.2508	93.01	0.1732	3.9558	125
126	0.9841	14.763	2.323	17.086	30.285	109.8	140.1	0.05821	0.1989	0.2571	94.01	0.1749	4.0649	126
127	1.016	14.788	2.401	17.189	30.526	113.4	143.9	0.05862	0.2050	0.2636	95.00	0.1766	4.1765	127
128	1.048	14.813	2.482	17.295	30.766	117.0	147.8	0.05903	0.2113	0.2703	96.00	0.1783	4.2907	128
129	1.082	14.839	2.565	17.404	31.007	120.8	151.8	0.05944	0.2178	0.2772	97.00	0.1800	4.4076	129
130	1.116	14.864	2.652	17.516	31.248	124.7	155.9	0.05985	0.2245	0.2844	98.00	0.1817	4.5272	130
131	1.152	14.889	2.742	17.631	31.489	128.8	160.3	0.06026	0.2314	0.2917	99.00	0.1834	4.6495	131
132	1.189	14.915	2.834	17.749	31.729	133.0	164.7	0.06067	0.2386	0.2993	100.00	0.1851	4.7747	132
133	1.227	14.940	2.930	17.870	31.970	137.3	169.3	0.06108	0.2459	0.3070	101.00	0.1868	4.9028	133
134	1.267	14.965	3.029	17.994	32.211	141.8	174.0	0.06148	0.2536	0.3151	102.00	0.1885	5.0337	134
135	1.308	14.990	3.132	18.122	32.452	146.4	178.9	0.06189	0.2614	0.3233	103.00	0.1902	5.1676	135
136	1.350	15.016	3.237	18.253	32.692	151.2	183.9	0.06229	0.2695	0.3318	104.00	0.1918	5.3046	136
137	1.393	15.041	3.348	18.389	32.933	156.1	189.0	0.06270	0.2778	0.3405	105.00	0.1935	5.4446	137
138	1.439	15.066	3.462	18.528	33.174	161.2	194.4	0.06310	0.2865	0.3496	106.00	0.1952	5.5878	138
139	1.485	15.091	3.580	18.671	33.414	166.5	199.9	0.06350	0.2954	0.3589	107.00	0.1969	5.7342	139

[a] Compiled by John A. Goff and S. Gratch.

TABLE 4—THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG)—*Continued*

Fahr. Temp. t(F)	Humidity Ratio W_s	Volume cu ft/lb dry air v_a	v_{as}	v_s	Enthalpy Btu/lb dry air h_a	h_{as}	h_s	Entropy Btu per (°F) (lb dry air) s_a	s_{as}	s_s	Condensed Water Enthalpy Btu/Lb h'_w	Entropy Btu/(Lb)(°F) s'_w	Vap. Press. In. Hg p_s	Fahr. Temp. t(F)
140	0.1534	15.117	3.702	18.819	33.655	172.0	205.7	0.06390	0.3047	0.3686	107.99	0.1985	5.8838	140
141	0.1584	15.142	3.829	18.971	33.896	177.7	211.6	0.06430	0.3142	0.3785	108.99	0.2002	6.0367	141
142	0.1636	15.167	3.961	19.128	34.136	183.6	217.7	0.06470	0.3241	0.3888	109.99	0.2018	6.1930	142
143	0.1689	15.192	4.098	19.290	34.377	189.7	224.1	0.06510	0.3343	0.3994	110.99	0.2035	6.3527	143
144	0.1745	15.218	4.239	19.457	34.618	196.0	230.6	0.06549	0.3449	0.4104	111.99	0.2051	6.5160	144
145	0.1803	15.243	4.386	19.629	34.859	202.5	237.4	0.06589	0.3559	0.4218	112.99	0.2068	6.6828	145
146	0.1862	15.268	4.539	19.807	35.099	209.3	244.4	0.06629	0.3672	0.4335	113.99	0.2084	6.8532	146
147	0.1924	15.293	4.698	19.991	35.340	216.4	251.7	0.06669	0.3790	0.4457	114.99	0.2101	7.0273	147
148	0.1989	15.319	4.862	20.181	35.581	223.7	259.3	0.06708	0.3912	0.4583	115.99	0.2117	7.2051	148
149	0.2055	15.344	5.033	20.377	35.822	231.3	267.1	0.06748	0.4038	0.4713	116.99	0.2134	7.3867	149
150	0.2125	15.369	5.211	20.580	36.063	239.2	275.3	0.06787	0.4169	0.4848	117.99	0.2150	7.5722	150
151	0.2197	15.394	5.396	20.790	36.304	247.3	283.6	0.06827	0.4304	0.4987	118.99	0.2167	7.7616	151
152	0.2271	15.420	5.587	21.007	36.545	255.9	292.4	0.06866	0.4445	0.5132	119.99	0.2183	7.9550	152
153	0.2349	15.445	5.788	21.233	36.785	264.7	301.5	0.06906	0.4591	0.5282	120.99	0.2200	8.1525	153
154	0.2430	15.470	5.996	21.466	37.026	273.9	310.9	0.06945	0.4743	0.5438	121.99	0.2216	8.3541	154
155	0.2514	15.496	6.213	21.709	37.267	283.5	320.8	0.06984	0.4901	0.5599	122.99	0.2232	8.5599	155
156	0.2602	15.521	6.439	21.960	37.508	293.5	331.0	0.07023	0.5066	0.5768	123.99	0.2248	8.7701	156
157	0.2693	15.546	6.675	22.221	37.749	303.9	341.7	0.07062	0.5237	0.5943	124.99	0.2265	8.9846	157
158	0.2788	15.571	6.922	22.493	37.990	314.7	352.7	0.07101	0.5415	0.6125	125.99	0.2281	9.2036	158
159	0.2887	15.597	7.178	22.775	38.231	326.0	364.2	0.07140	0.5600	0.6314	127.00	0.2297	9.4271	159
160	0.2990	15.622	7.446	23.068	38.472	337.8	376.3	0.07179	0.5793	0.6511	128.00	0.2313	9.6556	160
161	0.3098	15.647	7.727	23.374	38.713	350.1	388.8	0.07218	0.5994	0.6716	129.00	0.2329	9.8876	161
162	0.3211	15.672	8.020	23.692	38.954	363.0	402.0	0.07257	0.6204	0.6930	130.00	0.2345	10.125	162
163	0.3329	15.698	8.326	24.024	39.195	376.5	415.7	0.07296	0.6423	0.7153	131.00	0.2361	10.367	163
164	0.3452	15.723	8.648	24.371	39.436	390.5	429.9	0.07334	0.6652	0.7385	132.00	0.2377	10.614	164
165	0.3581	15.748	8.985	24.733	39.677	405.3	445.0	0.07373	0.6892	0.7629	133.00	0.2393	10.866	165
166	0.3716	15.773	9.339	25.112	39.918	420.8	460.7	0.07411	0.7142	0.7883	134.00	0.2409	11.123	166
167	0.3858	15.799	9.708	25.507	40.159	437.0	477.2	0.07450	0.7405	0.8150	135.01	0.2426	11.385	167
168	0.4007	15.824	10.098	25.922	40.400	454.0	494.4	0.07488	0.7680	0.8429	136.01	0.2441	11.652	168
169	0.4163	15.849	10.508	26.357	40.641	471.8	512.4	0.07527	0.7969	0.8722	137.01	0.2457	11.925	169
170	0.4327	15.874	10.938	26.812	40.882	490.6	531.5	0.07565	0.8273	0.9030	138.01	0.2473	12.203	170
171	0.4500	15.900	11.391	27.291	41.123	510.4	551.5	0.07603	0.8592	0.9352	139.01	0.2489	12.486	171
172	0.4682	15.925	11.870	27.795	41.364	531.3	572.7	0.07641	0.8927	0.9691	140.01	0.2505	12.775	172
173	0.4875	15.950	12.376	28.326	41.605	553.3	594.9	0.07680	0.9281	1.0049	141.01	0.2521	13.069	173
174	0.5078	15.975	12.911	28.886	41.846	576.5	618.3	0.07718	0.9654	1.0426	142.02	0.2537	13.369	174

[a] Compiled by John A. Goff and S. Gratch.

TABLE 4—THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG)—*Concluded*

Fahr. Temp. t(F)	Humidity	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
	Ratio W_s	v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h'_w	Entropy Btu/(Lb)(°F) s'_w	Vap. Press. In. Hg p_s	
175	0.5292	16.001	13.475	29.476	42.087	601.1	643.2	0.07756	1.005	1.083	143.02	0.2553	13.675	175
176	0.5519	16.026	14.074	30.100	42.328	627.1	669.4	0.07794	1.047	1.125	144.02	0.2568	13.987	176
177	0.5760	16.051	14.710	30.761	42.569	654.7	697.3	0.07832	1.091	1.169	145.02	0.2584	14.304	177
178	0.6016	16.076	15.386	31.462	42.810	684.1	726.9	0.07870	1.137	1.216	146.03	0.2600	14.628	178
179	0.6288	16.102	16.104	32.206	43.051	715.2	758.3	0.07908	1.187	1.266	147.03	0.2616	14.958	179
180	0.6578	16.127	16.870	32.997	43.292	748.5	791.8	0.07946	1.240	1.319	148.03	0.2631	15.294	180
181	0.6887	16.152	17.689	33.841	43.534	783.9	827.4	0.07984	1.296	1.376	149.03	0.2647	15.636	181
182	0.7218	16.177	18.565	34.742	43.775	821.9	865.7	0.08021	1.357	1.437	150.04	0.2662	15.985	182
183	0.7572	16.203	19.504	35.707	44.016	862.5	906.5	0.08059	1.421	1.502	151.04	0.2678	16.340	183
184	0.7953	16.228	20.513	36.741	44.257	906.2	950.5	0.08096	1.490	1.571	152.04	0.2693	16.702	184
185	0.8363	16.253	21.601	37.854	44.498	953.2	997.7	0.08134	1.565	1.646	153.05	0.2709	17.071	185
186	0.8805	16.278	22.775	39.053	44.740	1004	1049	0.08171	1.645	1.727	154.05	0.2724	17.446	186
187	0.9283	16.304	24.047	40.351	44.981	1059	1104	0.08208	1.731	1.813	155.05	0.2740	17.828	187
188	0.9802	16.329	25.427	41.756	45.222	1119	1164	0.08245	1.825	1.907	156.06	0.2755	18.217	188
189	1.037	16.354	26.934	43.288	45.463	1184	1229	0.08283	1.928	2.011	157.06	0.2771	18.614	189
190	1.099	16.379	28.580	44.959	45.704	1255	1301	0.08320	2.039	2.122	158.07	0.2786	19.017	190
191	1.166	16.405	30.385	46.790	45.946	1332	1378	0.08357	2.161	2.245	159.07	0.2802	19.427	191
192	1.241	16.430	32.375	48.805	46.187	1418	1464	0.08394	2.296	2.380	160.07	0.2817	19.845	192
193	1.324	16.455	34.581	51.036	46.428	1513	1559	0.08431	2.444	2.528	161.08	0.2833	20.271	193
194	1.416	16.480	37.036	53.516	46.670	1619	1666	0.08468	2.609	2.694	162.08	0.2848	20.704	194
195	1.519	16.506	39.785	56.291	46.911	1737	1784	0.08505	2.794	2.879	163.09	0.2864	21.145	195
196	1.635	16.531	42.885	59.416	47.153	1871	1918	0.08542	3.002	3.087	164.09	0.2879	21.594	196
197	1.767	16.556	46.402	62.958	47.394	2022	2069	0.08579	3.238	3.324	165.10	0.2895	22.050	197
198	1.917	16.581	50.426	67.007	47.636	2195	2243	0.08616	3.507	3.593	166.10	0.2910	22.514	198
199	2.091	16.607	55.074	71.681	47.877	2395	2443	0.08653	3.817	3.904	167.11	0.2925	22.987	199
200	2.295	16.632	60.510	77.142	48.119	2629	2677	0.08689	4.179	4.266	168.11	0.2940	23.468	200

[a] Compiled by John A. Goff and S. Gratch.

of saturated liquid. This and values at the higher temperatures have been computed from the integral of (9.1). All vapor pressure data have been converted to inches of mercury by means of the conversion factor 29.921 inches Hg/atm.

The next items entered in Table 4 are the specific enthalpy h_w' and specific entropy s_w' *of pure water at standard atmospheric pressure.* These were obtained by applying suitable corrections to the corresponding values at saturation which were calculated in the manner explained under the heading *Condensed Water.* The correction to the specific enthalpy is

$$\triangle h = \left[\frac{\partial (v\tau)}{\partial \tau}\right]_p (p - p_s) \qquad 17.1$$

that to the specific entropy is given by

$$T\triangle s = \triangle h - v\,(p - p_s) \qquad 17.2$$

and turns out to be entirely negligible. The decision to list values at standard atmospheric pressure rather than at saturation pressure has been based on consideration of the practical uses to which the table is most likely to be put. Thus, when moist air is cooled below the dew-point, liquid (or ice) condenses out under the observed or *total* pressure p. Strictly speaking, the condensed phase is not pure water but contains a small quantity of dissolved air which affects its enthalpy and entropy slightly; however, corrections for this effect have not been applied.

The next item entered in the table is the humidity ratio at saturation W_s. The corresponding mol fraction of dry air x_s is computed directly from (14.6) and then converted to humidity ratio by the definition,

$$W_s = 18.0154\,(1 - x_s)/28.966\,x_s \qquad 17.3$$

The specific volume v_a, specific enthalpy h_a, and specific entropy s_a of dry air are computed from (2.0), (3.0), (3.1), respectively, using (2.4) for numerical values of the second virial coefficient A_{aa} and its derivatives B_{aa} and C_{aa} and completely ignoring the third virial coefficient A_{aaa}. That (2.4) is sufficiently accurate to stand the differentiations necessary to obtain values of B_{aa} and C_{aa} from it has been tested by Bridgeman,[5] using Joule-Thomson and other calorimetric data for comparison. The method of obtaining zero-pressure enthalpies and reduced entropies has been explained in Section I, Enthalpy and Entropy.

The saturation volume v_s, saturation enthalpy h_s, and saturation entropy s_s are computed from (13.8), (13.9), (13.10), respectively, using for mol fraction $1\text{-}x_s$ the values computed from (14.6) and dividing by 28.966 x_s in order to refer to unit weight of dry air. The determination of the various temperature functions appearing in these equations has been explained in considerable detail.

The humidity ratio W, that is the weight of water vapor per unit weight of dry air, may have any value between zero and W_s, the humidity ratio at saturation. The ratio $\mu = W/W_s$, called *degree of saturation* or *per cent*

saturation, may therefore have any value between zero and unity. As a close approximation the corresponding values of volume v, enthalpy h, and entropy s, may be computed from the linear equations,

$$\left.\begin{aligned} v &= v_a + \mu(v_s - v_a) \\ h &= h_a + \mu(h_s - h_a) \\ s &= s_a + \mu(s_s - s_a) \end{aligned}\right\} \qquad 17.4$$

To facilitate such computations, the differences appearing in parentheses in (17.4) have been included in Table 4 under the headings v_{as}, h_{as} and s_{as}, respectively.

The use of linear interpolation according to (17.4) involves some loss of accuracy, especially at the higher temperatures, but hardly enough to justify complicating the tables to regain it. When it comes to representing Table 4 in the form of a Mollier Chart, the actual isotherm curvature can be incorporated without introducing complications except in the actual drawing of the chart.

V. CONCLUSION

From the discussion of preceding sections it is clear that there is still room for improvement in the primary data which constitute the ingredients of the present formulation. This is especially true of the virial and interaction coefficients. Also there is much room for improvement in the theory by which these primary data are put together as suggested by the arbitrary manner in which the third virial coefficients have been handled.

In spite of the above shortcomings we believe that the present formulation is more than sufficiently accurate for all practical purposes now and in the near future. All available data have been represented well within their respective accuracies and only in a few instances have possible refinements been neglected. For example, to repeat the state-sum calculation of the zero-pressure properties using the newest and best spectroscopic data would improve their accuracy somewhat, but such improvement would not be noticed in the final formulation because of irreducible inaccuracies in other data used.

The only part of the present formulation which is offered with reservations other than those with which any numerical computations must be offered is the part at higher temperatures, say above 150 F. Here it has been necessary to exercise arbitrariness in handling the third virial and corresponding interaction coefficients, and it may be that some error has been introduced. However, without further information, both experimental and theoretical, the authors know of no better procedure to follow. To remove this and other sources of uncertainty would probably require an elaborate research program similar to the steam research program of the past two decades. But it is to be doubted that such additional experimentation is warranted by actual needs, at any rate, now and for some time to come.

Although Table 4 is for standard atmospheric pressure only, the necessary information is at hand for the preparation of similar tables for other pressures. Accurate tables can be prepared for fairly high pressures if there is a real need for them, though as pressure increases the effects of the third virial

coefficient become more pronounced. At any rate, suggestions regarding needs for tables at other pressures are invited.

ACKNOWLEDGMENTS

It is gratifying to note that the Council of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has recently agreed to sponsor the formation of an Inter-Society Committee to attempt to reach agreement on a standard table of properties to be recommended for general adoption.

The authors have derived a great amount of pleasure from the moist air research reported herein and for the opportunity to engage in it wish to thank the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, who have generously supported the program. It is also a pleasure to acknowledge indebtedness to Dr. A. C. Bates, formerly Assistant Professor in Mechanical Engineering, University of Pennsylvania, and J. R. Andersen, Instructor in Mechanical Engineering, University of Pennsylvania, for their contributions, both experimental and analytical. Dr. H. Margenau, Sloane Laboratory of Physics, Yale University, has given generously of advice and guidance in the use of modern physical theory in the analytical part of the investigation. A special fund for research in Mechanical Engineering provided by Col. E. M. Chance, President, United Engineers and Constructors, has made possible the purchase of a modern electric calculator, without which the labor involved in necessary routine calculations would have been greatly increased.

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DISCUSSION

W. L. FLEISHER, New York, N. Y.: I am very much interested in talking about this paper because I think I was Chairman of the Committee on Research at the time the question of sponsoring this work came up. I do not mind saying that I had some difficulty getting the Committee on Research to appropriate money to carry on anything that was as abstruse and as scientific as the work that Dean Goff intended to pursue, but I think this paper is really a transitional paper as far as the Society is concerned. It is the finest fundamental paper that has been developed by the Society, and I think that probably the work done, although it may have no practical significance from the standpoint of a great many members, represents the objective which I always wanted to see the Society develop. In other words, it gives us such a standing as a scientific Society as we have never had at any other time. The Society, as everybody knows, has been much more of a practical organization than a scientific organization in the minds of most people, and the prestige that a paper of this kind will bring us is very desirable because in a way we have done something that even the *American Society of Mechanical Engineers*, of which I have been a member for many years, has not accomplished.

Consequently, although probably most of us will not understand a good bit of the mathematics and theory of this paper, its significance is so great that I think we owe a debt of gratitude to Dean Goff for having contributed to the Society something that any of the other scientific societies would give a great deal to have presented to them.

JOHN JAMES, Cleveland, Ohio: The true significance of the results of this paper can hardly be appreciated by those privileged to hear the presentation today. However, in the years to come as the Society is gradually recognized for its foresight in

initiating and sponsoring this research work it is conceivable that Dean Goff's results will attain the same standing as the well-known steam tables.

At the present time there is great confusion in our industry regarding the proper data to use when performing precise engineering computations and for that reason the work of the new Inter-Society Committee on psychrometry should be of inestimable value.

Many engineers were educated to use the term *total heat* in determining the amount of energy in air and water mixtures. A short time ago, the phrase *heat content* was introduced, and recently the term *sigma function* was presented, all of which, along with the question as to whether the *heat of the liquid* should be added, have confused the situation in the minds of most individuals. *Enthalpy,* the term included in this paper, is now recognized as the true value to use.

Some engineers are prone to question the practical significance of these data as applied to field problems with the contention that the small differences are not recognizable in the final result. This is true to a certain extent but this does not discount the value of having thermodynamically consistent data which may be considered as standard by recognized authorities.

One of the major difficulties in using psychrometric data in the past has been that our instruments for measuring dry-, wet- and dew-point temperatures have not been accurate enough to warrant referring the results to tabular data possessing extreme accuracy. Fortunately progress is also being made in this field and it is hoped and expected that a greatly improved instrument will be made available for accurately measuring the thermal conditions of an environment.

It seems to me that this cooperative research work with the University of Pennsylvania has been of outstanding value and I think we all owe a debt of gratitude to the author.

CYRIL TASKER, Cleveland, Ohio: It would be impertinent of me to pay a tribute to Dean Goff and his work after the tributes that have been paid. He not only has been doing a magnificent job for the Society with this fundamental work, but he has been a very valued member of the Committee on Research and you will be pleased to know that he is going to be a member for a further three-year term. I think that fact indicates our belief in his work and the tremendous assistance he has given to the Society.

Coupled with this fundamental work is the problem upon which Mr. James lightly touched, namely, the problem of instrumentation. This problem is one which is very prominently before the Committee on Research, and the Committee on Psychrometry, of which J. H. Walker is chairman. We realize very definitely that having these fundamental data on a sound basis, our next step is to see how we can use them for our day to day operations. Immediately we come up against the fact that our instruments are by no means as accurate as our theory, and therefore the Research Laboratory has taken a step toward bringing into being a program of work on psychrometric instrumentation.

We arranged for a preliminary report to be made on the present methods of measuring the properties of mixtures of air and water vapor. That report has been mimeographed and is available at the Research Laboratory. It is by no means, and it was not intended to be, the last word on the subject. It was not written as a textbook on the subject, but we have tried to indicate in it the main errors that one might encounter in using the instruments which are available today for our ordinary work—what to watch for; how good some of them are; how poor some of them are; and where possible improvements might be made. If you wish to have a copy of this mimeographed report, please write to the Laboratory and we will be happy to send it to you. In sending it to you we are giving something away; at the same time we are asking something. When you get that report we hope that you will read it and tell us what you think about it quite frankly. If you do not agree with the statements that are made therein, we want to know about it, because, to come back to

my standard slogan on this work, all this is your program and we want to know what your thoughts are on this question of instrumentation and the difficulties you have run into, so that we can set up a program of experimental work to help solve some of those difficulties for you.

We are also hoping to institute in the very near future a program of experimental work on instrumentation, and, particularly at the present time, on determinations of the true dew-point.

MR. FLEISHER: In reading this paper over in advance I was rather confused by the change from negative to positive influence of the author's virial coefficients. It would seem that the virial coefficients have a negative value up to a certain point and then change to a positive, and I would appreciate it if Dean Goff would explain this to me.

C. O. MACKEY, Ithaca, N. Y.: I think this is a highly commendable paper and a very important project. There have been a great many people who have prepared tables of thermodynamic properties of mixtures of air and water vapor. These tables are not in agreement at the present time and there has been much duplication of effort. Anything that we can do to get a set of tables which will be generally adopted even nationally, if not internationally, is an extremely important forward step. I do have several specific suggestions concerning the tables which I hope will be considered merely as suggestions, and my purpose is to try to secure a more general adoption by engineers of these tables by changing their form, perhaps.

These specific suggestions are: *First,* I would like to see a more complete, condensed separation of the actual equations used in the preparation of the tables from the justification for the formulas and the constants in those equations. *Second,* I would like to see a complete separation of the table of properties of moist air. *Third,* I would like to see an expansion of the methods used in preparing the tables to find the thermodynamic properties of moist air when the atmospheric pressure is different from the standard atmospheric pressure of 29.921 in. Hg, and the same suggestion would apply to finding the properties of dry air. The properties of dry air alone are sufficiently important so that Professor Keenan at M.I.T. is preparing a book largely on the properties of dry air.

In illustration of what I mean by the effect of departures from the standard pressure, considering just the properties of dry air—and that is as far as I have been able to go in looking over the results—if the standard total pressure is used at a temperature of 70 F the specific volume of dry air is, according to the table, 13.349 cu ft per pound. If the pressure changes to 28.5 in. Hg, that volume becomes 14.015 cu ft per pound.

It happens that the specific volume is made up of the algebraic sum of two functions: one, a pressure function, and the other a temperature function. The second virial coefficient is the temperature function alone, and if the sum is separated into two parts in the table, only one part of the sum needs to be corrected for departures from the standard pressure. The pressure function can be easily corrected by multiplying or dividing by pressure ratios, as the case may be, and then this part may be added algebraically to the second virial coefficient, which is independent of pressure —and so on through the other properties, enthalpy and entropy.

To illustrate what is meant by these suggestions, condensed equations and a table of the thermodynamic properties of dry air are included. These equations may not agree precisely with those of the author's, for they involve fitting an empirical equation to the specific heat at constant zero pressure of dry air.

Equations for the Thermodynamic Properties of Dry Air at Any Atmospheric Pressure of p in. Hg abs.:

v_a = specific volume, cu ft/lb;
t = temperature, deg F;
T = temperature, deg $R = t + 459.7$;
p = atmospheric pressure, in. Hg abs.;

A_{aa} = second virial coefficient, cu ft/lb;
h_a = specific enthalpy, Btu/lb
S_a = specific entropy, Btu/lb R } datum state: $t = 0$ F; $p = 29.921$ in. Hg abs
$C_{p_a}^{(0)}$ = specific heat at constant zero pressure, Btu/lb F.

$$v_a = v_1 - A_{aa} \qquad (1)$$

$$v_1 = \frac{0.7543\ T}{p} \qquad (2)$$

$$A_{aa} = -0.0225 + \frac{13.056}{T} + \frac{3.87\ (10)^5}{T^2} \qquad (3)$$

$$h_a = h_1 + 0.137 + \int_{459.7}^{T} C_{p_a}^{(0)}\, dT \qquad (4)$$

$$h_1 = p\left[0.002046 - \frac{2.3736}{T} - \frac{1.4067\ (10)^6}{T^3}\right] \qquad (5)$$

$$C_{p_a}^{(0)} = 0.23975 - 4.4066\ (10)^{-6}\ T + 9.2226\ (10)^{-9}\ T^2 \qquad (6)$$

$$S_a = S_1 + S_2 + 0.52522 + \int_{459.7}^{T} \frac{C_{p_a}^{(0)}\, dT}{T} \qquad (7)$$

$$S_1 = -0.15447\ \ln p \qquad (8)$$

$$S_2 = -p\left[\frac{1.1868}{T^2} + \frac{1.0550\ (10)^5}{T^4}\right] \qquad (9)$$

I do not think we should lose sight of the fact that this paper is only a beginning and two of the previous speakers, Mr. James and Mr. Tasker, have emphasized that point. We shall need additional tables for the properties of moist air to be used when the total pressure, the dry-bulb temperature and the thermodynamic wet-bulb temperature are known, but when the state is not the saturated state, when the relative humidity is less than 100 per cent.

In addition we shall need a study of the experimental problem of determining the state of a mixture of air and water vapor.

In other words, to put the problem as a question: What is the thermodynamic wet-bulb temperature or the temperature of adiabatic saturation corresponding to a wet-bulb temperature observed with, say, a wet bulb unshielded from the effect of radiation, when we know the air velocity or air movement past that wetted bulb?

The engineer will ultimately want tables or charts which permit finding the three properties of state—specific volume, specific entropy, and specific enthalpy—when the dry-bulb temperature, thermodynamic wet-bulb temperature, and total pressure of the mixture are known. For example, what are these three properties when the dry-bulb temperature is 95 F, the thermodynamic wet-bulb temperature is 75 F, and the total pressure of the mixture is 29.00 in. Hg abs?

I emphasize again that this is just a beginning but a very important first step and a very commendable first step in an extremely large problem.

CHAIRMAN WINSLOW: I know we all concur with Mr. Fleisher's tribute and the tribute of the other speakers to Dean Goff. I do want to take a little exception to a certain suggestion, however, in Mr. Fleisher's remarks, implying, as it seemed to me, a defensive attitude with regard to our Society. I would like to call your attention to this fact: mechanical engineering, electrical engineering and civil engineering are older sciences and naturally more has been done of a fundamental type in universities and research institutions, but from the standpoint of the Society, as a professional society, I think we can challenge any other professional society of any kind

with which I am familiar to show that it has contributed one tenth as much as this Society has contributed, as a society, to the advancement of the science with which it deals.

AUTHOR'S CLOSURE: I very much appreciate the comments of Messrs. Fleisher, James, Tasker and Professor Mackay. They give me a great deal of satisfaction. Of course the problem itself has been a very interesting one and I appreciate the opportunity which the Society has given us of working on it; because it has opened the door to other problems where the ideas which we have accumulated from this one will, I think, be very helpful.

Our understanding of thermodynamics has been greatly enhanced by the steam research program of the past 20 years. This program was undertaken largely for the

SKELETON TABLES OF THE THERMODYNAMIC PROPERTIES OF
OF 29.921 IN.

(With instructions for finding these properties at

TEMP.		SECOND VIRIAL COEFF.	SPECIFIC VOLUME			SPECIFIC ENTHALPY
t	v_1	A_{aa}	$v_a = v_1 - A_{aa}$	h_1	$0.137 + \int_{459.7}^{T} C_{p_a}^{(0)}\, dT$	$h_a = h_1 + 0.137 + \int_{459.7}^{T} C_{p_a}^{(0)} dT$
F	ft³/lb	ft³/lb	ft³/lb	Btu/lb	Btu/lb	Btu/lb
−160	7.5556	0.0354	7.520	−0.332	−38.172	−38.504
−150	7.8077	0.0327	7.775	−0.310	−35.778	−36.088
70	13.3540	0.0047	13.349	−0.117	16.933	16.816
80	13.6061	0.0041	13.602	−0.097	19.318	19.221
190	16.3793	−0.0010	16.380	−0.063	45.767	45.704
200	16.6314	−0.0014	16.633	−0.061	48.180	48.119

[a] For an atmospheric pressure of p in. Hg abs. instead of 29.921 in. Hg abs.

$$v_a = \frac{29.921}{p} v_1 - A_{aa};$$

$$h_a = \frac{p}{29.921} h_1 + 0.137 + \int_{459.7}^{T} C_{p_a}^{(0)}\, dT;$$

$$S_a = -0.15447 \ln p + \frac{p}{29.921} S_1 + 0.52522 + \int_{459.7}^{T} \frac{C_{p_a}^{(0)}\, dT}{T}$$

purpose of reaching agreement upon standard properties of steam. It would have been possible to say "if it is just a matter of standardization why not follow the suggestion of Dr. Callendar, English physicist, and pick out something simple as a basis for agreement being willing to sacrifice accuracy, if necessary, in order to gain simplicity." If we had adopted such a procedure I am sure it would have been shortsighted; because we should not have gained from it the understanding of thermodynamics which we obtained by placing emphasis on accuracy. I believe that the same philosophy should guide our efforts to reach agreement on standard properties of moist air.

One thing in particular that we learned through steam research was the importance of paying close attention to the situation at zero pressure where, paradoxical as it

may seem, our knowledge of the properties of gases and vapors is more precise than at any accessible pressure. Modern developments in the science of interpreting spectra of gases have enabled us to compute zero-pressure values of specific heat, enthalpy, and entropy which meet all present day requirements of accuracy, at any rate, in the case of simple molecules.

Having assembled accurate information at zero pressure, the next problem is to move off toward finite pressures. At present this departure has to be made largely on the basis of experimental data; but again, as developments in modern physics come along, we may expect to make the departure on the basis of theoretical considerations. At the present time these considerations are used mainly to correlate existing experimental data.

DRY AIR AT THE STANDARD ATMOSPHERIC PRESSURE
HG ABS.

any atmospheric pressure of p in. Hg abs.[a])

			SPECIFIC ENTROPY	TEMP.
S_1	S_2	$0.52522 + \int_{459.7}^{T} \frac{C_{p_a}^{(0)}\,dT}{T}$	$S_a = S_1 + S_2 + 0.52522 + \int_{459.7}^{T} \frac{C_{p_a}^{(0)}\,dT}{T}$	
Btu/lb R	Btu/lb R	Btu/lb R	Btu/lb R	F
−0.52498	−0.00079	0.42277	−0.10300	−160
−0.52498	−0.00071	0.43061	−0.09508	−150
−0.52498	−0.00017	0.55920	0.03405	70
−0.52498	−0.00016	0.56368	0.03854	80
−0.52498	−0.00010	0.60828	0.08320	190
−0.52498	−0.00010	0.61197	0.08689	200

Example: $t = 70$ F; $p = 28.50$ in. Hg abs.:

$$v_a = \frac{29.921}{28.50}(13.3540) - 0.0047 = 14.0198 - 0.0047 = 14.015 \text{ ft}^3/\text{lb};$$

$$h_a = \frac{28.50}{29.921}(-0.117) + 16.933 = -0.111 + 16.933 = 16.822 \text{ Btu/lb};$$

$$\begin{aligned} S_a &= -0.15447 \ln 28.5 + \frac{28.5}{29.921}(-0.00017) + 0.55920 \\ &= -0.51746 - 0.00016 + 0.55920 \\ &= 0.04158 \text{ Btu/lb R} \end{aligned}$$

In answer to Mr. Fleisher's questions, I would refer to Fig. 3, which gives the so-called law of force between a pair of interacting molecules. Incidentally, it is interesting to note that these factors, which may be either attractive or repulsive depending on the distance of separation, are purely electrostatic. Each molecule has a certain distribution of electric charges on its constituent atoms and electrons. As these molecules move around they produce changes in the electric field and thereby interact with other rotating molecules. The problem then is to find out how the electrostatic potential of a pair of rotating molecules depends upon their mutual orientation and on their distances of separation. When this is known the methods of statistical mechanics enable us to compute the net effect and thereby predict the second coefficient. It turns out that at high temperature where the translational motion of a

molecule becomes more and more rapid, they come within close distances of one another more and more often. But at close distances the force is repulsive and when repulsive forces predominate the second virial coefficient is negative and the specific volume at a given pressure exceeds that which would be predicted by the perfect gas laws.

On the other hand, at low temperatures where the translatory motion is relatively slow, molecules come close to one another relatively infrequently and the long-range attractive forces predominate. Therefore, at low temperatures the second virial coefficient is positive and the specific volume at a given pressure is less than what would be predicted by the perfect gas loss.

I believe Professor Mackey's first suggestion, that working formulæ on which the calculations of the table are based should be separated from the argument leading to them and reduced to convenient empirical form, should be given consideration.

Regarding a separate table giving the properties of dry air, this can be readily provided, at any rate for the range of temperature and pressure considered here, if it is found to be desirable.

With regard to tables for different pressures, we are in a position to extend the present table for standard atmospheric pressure either to higher or to lower pressures. We have stopped with the standard atmospheric pressure table mainly to wait until we have agreement on what range of pressures should be contemplated, what intervals of pressure should be chosen, etc.

The further suggestion from Professor Mackey that we might have, for instance, one table at zero pressure plus a series of tables giving corrections for higher pressures might be worked out. I am not sure that this arrangement will be a very satisfactory one but it is a worthwhile suggestion and perhaps the Technical Advisory Committee on Psychrometry will want to give it careful consideration.

Information regarding unsaturated mixtures, those for which the weight of water per pound of dry air is less than that called for by saturation, is all contained in the tables. You will note, for example, in the columns under volume that the lefthand column gives the specific volume of dry air. The righthand column gives the volume of the saturated mixture per pound of dry air, and the middle column gives the difference. In order to find the volume per pound of dry air of a 50 per cent saturated mixture, take the specific volume of dry air and add 50 per cent of the difference. This is exactly what is done in the case of steam. Thus in a mixture of liquid and vapor having a quality of 50 per cent take the specific volume of the liquid and add 50 per cent of the difference. The method is exact in the case of steam because it is a pure substance. The method is only approximate in the case of moist air. At temperatures above 100 or 150 F the loss of accuracy is somewhat larger than the uncertainty in the tabulated values; but below 100 F the loss of accuracy is probably negligible. Even for temperatures above 100 F no revision of the fundamental data is called for; merely a more elaborate method of presenting them.

1271

No. 1271

AIR COOLING COIL PROBLEMS AND THEIR SOLUTIONS

By Lawrence G. Seigel,* Cleveland, Ohio

This paper is the result of research work at Case School of Applied Science, at the instigation of the Bureau of Ships, Navy Department and in cooperation with the American Society of Heating and Ventilating Engineers.

COOLING coil problems have been solved, for the most part, by overall performance factors patterned after those in use by the heating industry for the purpose of rating steam coils and unit heaters. This method of coil rating has proved fairly satisfactory for use with air cooling in which no dehumidification takes place, but when moisture from the air stream is deposited on a coil surface, the use of overall coefficients is either inaccurate or becomes involved. A method based upon separate (or film) coefficients for air-side and refrigerant-side surface is offered here as a practical solution to the coil rating and performance problem.

Introduction

This method has been developed during a ten-year program of research at Case School of Applied Science and is based on the results of hundreds of tests of many different coil designs and on coils using both direct expansion and non-volatile refrigerants. It provides a simple means for obtaining manufacturers' rating data which requires only a few tests for each design of coil, and it affords a basis of calculation of coil performance which gives direct results for any desired conditions without trial and error and without the use of many pages of tables that involve tedious interpolation.

The basis for this *Humidity Method* was presented to the A.S.H.V.E. in a paper in 1938.[1] At that time, however, the method had not been fully developed, and the data reported were the results of tests on water coils only. As several hundred additional tests have been run it is the purpose of this paper to demonstrate how the method previously outlined has been simplified, and how it can be applied to either the rating or the selection of any type of finned coil operating under conditions of forced convection.

Coil Performances

In the paper already mentioned, a mathematical derivation was presented which defined the locus of exit air conditions for a coil operating with decreas-

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[1] Performance of Surface-Coil Dehumidifiers for Comfort Air Conditioning, by G. L. Tuve and L. G. Seigel. (A.S.H.V.E. Transactions, Vol. 44, 1938, p. 523.)

Presented at the 51st Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, Mass., January, 1945.

ing surface temperature and constant entering air conditions. This locus is represented by the line 1-2-3-4 in Fig. 1, which has been drawn for a typical coil operating at a given face velocity. A point on line 1-2 represents dry cooling only, and a point on the line 2-3-4 indicates that dehumidification has taken place, as well as dry cooling. A line through 1-3 or 1-4 will be recognized as the commonly known load ratio line, while the curved line 1-M represents the condition curve for a particular condition of operation. It should be understood that the line 1-2-3-4 is not intended to indicate anything about the condi-

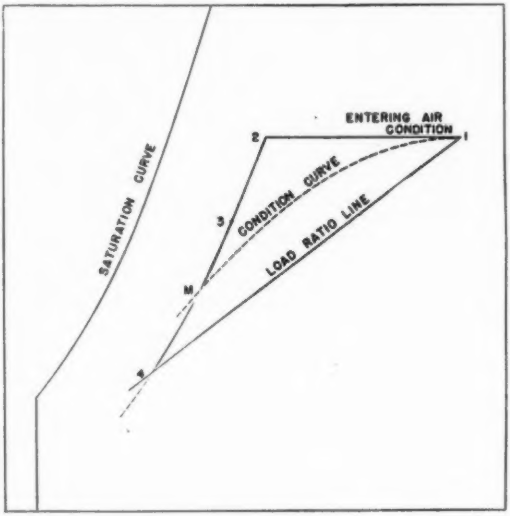


FIG. 1. PSYCHROMETRIC LAYOUT SHOWING COIL PERFORMANCE

tion of the air as it passes *through* the coil from row to row. It simply represents on the psychrometric chart the possible location of the condition of the air as it *leaves* a coil operating with decreasing refrigerant (or surface) temperature. Condition curves similar to 1-M can be constructed through the use of the *Humidity Method* if they are desired, and for purposes of illustration Figs. 2 and 3 have been prepared. These show a comparison of condition curves determined by Goodman's method and by the *Humidity Method*. The test data and coefficients used were obtained by Goodman.[2] In one case, as shown in Fig. 3, the coil surface was only partially wet and still fairly good agreement is demonstrated. However, these curves are usually not required in practice since the main concern is not what happens to the air as it passes through the coil, but what its condition is *leaving* the coil.

[2] Dehumidification of Air with Coils, by William Goodman. (*Refrigerating Engineering*, October, 1936.)

The line 2-3-4 in Fig. 1 has been established as a line at a constant horizontal distance from the saturation curve on the psychrometric chart. This line has been established mathematically in the 1938 paper [3] and data from many actual tests are available to provide further proof. Fig. 4 shows the results of one such test. To obtain the data of Fig. 4, air at a condition represented by point A was constantly supplied to a four row coil and the coil surface temperature was lowered by reducing the refrigerant temperature in such steps as were required to obtain points B through G. At point G, the refrigerant temperature was below the freezing point of water, and the test

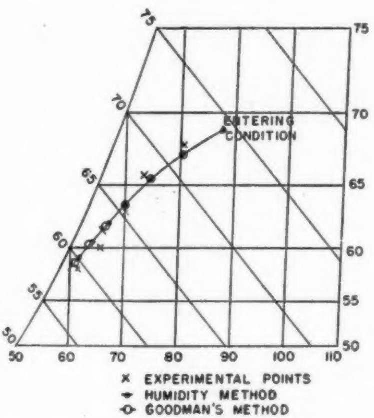


Fig. 2. Comparison of Methods for Determining Condition Curves

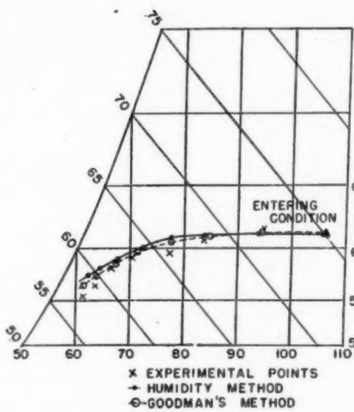


Fig. 3. Comparison of Methods for Determining Condition Curves

was stopped to prevent frosting of the coil. However, it is doubtful if a coil operating with a frosted surface would produce air any closer to the saturation curve than indicated by the points of Fig. 4. In support of this statement, the results of a test made with frosted coil surface are offered in Fig. 5. Although the points are not so consistent as those in Fig. 4 (due to the difficulty of testing under frosted conditions), it is still evident that the exit conditions do not approach the saturation curve. Many tests like that shown in Fig. 2 have been made. Therefore, in view of the mathematical derivation and the supporting experimental evidence, it seems safely established that the locus of the exit air conditions for a coil operating with constant entering conditions and decreasing refrigerant temperature is a line similar to line 1-2-3-4 of Fig. 1. Actually, when the exit conditions for a coil are partially located by the use of such a line, every coil problem (whether moisture is condensed or not) becomes a dry cooling problem which involves only sensible heat coefficients. Hence, the problem is not complicated by wet coil multipliers or the

[3] Loc. Cit. Note 1.

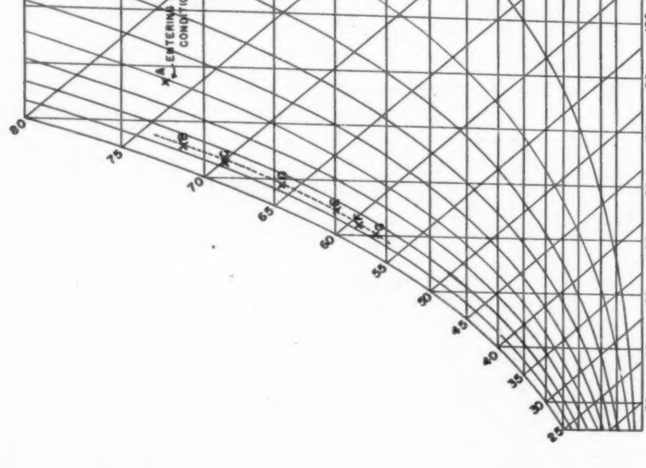


FIG. 4. AIR CONDITIONS LEAVING COIL OPERATING AT CONSTANT ENTERING CONDITION AND DECREASING REFRIGERANT TEMPERATURE

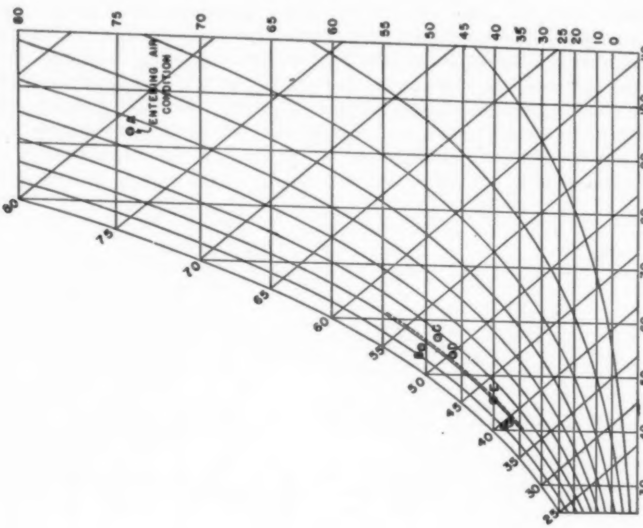


FIG. 5. AIR CONDITIONS LEAVING COIL OPERATING AT CONSTANT ENTERING CONDITION AND DECREASING REFRIGERANT TEMPERATURE AND FROSTED SURFACE

use of different coefficients for wet and dry surfaces, or by the determination of the relative portions of wet and dry surface.

It will be noted that mention of refrigerant temperature has been omitted from the foregoing discussion. This is because the performance of a coil is dependent upon air-side characteristics and is not affected by refrigerant-side variables except as these variables cause changes of surface temperature. In other words, it is possible to select the proper coil for a given job without reference to the refrigerant-side at all. Then it is only a matter of providing and controlling the proper refrigerant capacity to obtain the desired result. This is a definite advantage since air-side coefficients may be fairly well established while refrigerant coefficients may be subject to much variation. That is to say, the proper coil may be selected in any case from air-side data and any errors that result in the compressor selection (due to inaccurate data on refrigerant coefficients) may be corrected relatively easily by changing the compressor speed. This is much better than having a possible error carried over to the air-side as it would be by any method based on overall coefficients.

COIL CALCULATION EQUATIONS

In the use and application of the *Humidity Method,* only three simple equations and a psychrometric chart are necessary. These equations which are given in Table 1 involve the knowledge of two surface coefficients which must be obtained from test. A discussion of how these coefficients may be obtained follows, and samples of application of the equations will be given in the Appendix.

LIST OF SYMBOLS

A_s = Air-side surface area, sq ft/sq ft of coil face area per row of coil depth
C = A constant determined by experiment.
DP_1 = Dew-point temperature of air entering coil deg F.
DP_2 = Dew-point temperature of air leaving coil deg F.
N = Number of rows of coil depth.
Q_t = Total coil load Btu/hr/sq ft of coil face area.
R = Ratio, $\frac{T_1 - DP_1}{T_2 - DP_2}$
R_s = Ratio of external surface area to internal surface area.
T_1 = Dry-bulb temperature of air entering coil.
T_2 = Dry-bulb temperature of air leaving coil.
T_r = Refrigerant temperature deg F.
T_s = Coil surface temperature deg F.
V = Air velocity at coil face, ft/min.
G_a = Weight of air-vapor mixture lb/hr/sq ft of coil face area.
G_{da} = Weight of dry air lb/hr/sq ft of coil face area.
h_1 = Enthalpy of air-vapor mixture entering coil, Btu/lb of dry air.
h_2 = Enthalpy of air-vapor mixture leaving coil, Btu/lb of dry air.
h_a = Air-side film coefficient, Btu/hr/sq ft /deg F.
h_r = Refrigerant film coefficient, Btu/hr/sq ft/deg F.
n = A constant determined by experiment.

SURFACE COEFFICIENTS

Air-Side Coefficients: Since heat is transferred from air to a cooling coil by means of a difference in temperature between the air and the coil surface, it is clear that the performance of the coil is dependent upon the air-side characteristics of the coil only and is not affected by the type or kind of refrigerant used except as the refrigerant affects the surface temperature of the coil.

TABLE 1—COIL CALCULATION EQUATIONS

No.	Equation	Normal Use
1.	$\frac{h_a A_s N}{0.243\, G_a} = \log_e \frac{(T_1 - DP_1)}{(T_2 - DP_2)} = \log_e R$	Exit cond. location
2.	$T_s = \frac{(RT_2 - T_1)}{(R - 1)}$	Surface temperature
3.	$Q_t = \frac{h_r}{R_s} A_s N (T_s - T_r)$	Refrigerant temperature

These air-side characteristics then should be obtained most satisfactorily by use of air-side data alone. This may be done by application of Equation 1 to the results of wet coil tests, solving for h_a. If values of h_a obtained by use of this equation are then plotted against air velocity, the air-side coefficient will be evaluated in a form which may be expressed by the equation $h_a = CV^n$.

Many other ways of finding these coefficients have been suggested. Some of these have been tried for comparison purposes using the same test data for

TABLE 2—COMPARISON OF AIR-SIDE COEFFICIENTS
(*Calculated by three different methods*)

TEST	TOTAL LOAD BTU/HR	AIR FLOW LB/HR	AIR-SIDE COEFFICIENT h_a		
			Goodman Method	BCMI Method	Humidity Method
31	92,900	8760	18.65	19.10	18.42
32	80,750	6600	13.22	12.71	12.85
33	64,000	4590	9.38	9.04	9.03
34	96,200	8650	16.85	15.00	15.78
37	77,650	4650	9.92	9.60	9.35
38	121,400	10800	19.18	15.40	19.26

the same coil. The results are indicated in Table 2, which shows serious disagreement in only one case. Therefore, it is evident that air-side film coefficients may be obtained in various ways, all of which give approximately the same result. However, of all the methods tried, the *Humidity Method* required the fewest and simplest tests and the least complex calculations. The methods of Table 2 are as follows: (1) Goodman Method,[4] (2) Variation of BCMI method,[5] and (3) Humidity Method.

[4] Loc. Cit. Note 2.
[5] Proposed Standard Code for Testing Air Cooling Coils, *Blast Coil Manufacturer's Institute,* 1942.

To obtain surface coefficients by the *Humidity Method,* only three wet coil tests are required. However, this is the minimum number, and for the purpose of checking, it is recommended that at least six be run. The only requirements are that the coil be operating under dehumidifying conditions, and the air velocity be varied by at least three values in the range in which the results are to be applied. It does not matter if the surface of the coil is entirely wet or only partially wet. However, extremely light latent loads are very difficult to determine, and therefore it is recommended that tests having sensible to total load ratios of less than 70 per cent be used. To evaluate the coefficients from the test data, it is only necessary to solve Equation 1 for h_a

Table 3—Comparison of Refrigerant-Side Coefficients

(*Calculated by two different methods*)

Test	Total Load Btu/hr	Air Flow lb/hr	Refrigerant Coefficient h_r	
			Goodman Method	Humidity Method
31	92,900	8760	235	332
32	80,750	6600	341	315
33	64,000	4590	360	321
34	96,200	8650	287	356
37	77,650	4650	300	378
38	121,400	10800	316	390

and then plot this coefficient against the air velocity which prevailed during the test. If log-log paper is used for this plot, the result should be a straight line.

Refrigerant-Side Coefficients: The refrigerant-side coefficients required by Equation 3 may be evaluated from the same test data that were used for the determination of the air-side coefficients simply by the application of Equations 2 and 3. These coefficients will be quite erratic because of many factors such as variations in oil quantity, mean temperature difference, and refrigerant properties. To illustrate this variation and to demonstrate that it is independent of method, Table 3 has been prepared to show refrigerant coefficients calculated by two different methods for the same test data. The fact that the same proportional variation does not occur in the results of these methods probably indicates that a large part of the error is due to differences in determining the mean temperature difference between the surface and the refrigerant. More research is required on this subject and one project is already started under the sponsorship of the A.S.H.V.E. However, in spite of the erratic nature of these refrigerant coefficients, serious errors in coil selection will not result since the selection of the coil itself is based on air-side data. The selection of the refrigerant compressor may be slightly in error, but this will seldom be serious if the coefficients were evaluated under typical operating conditions.

TABLE 4—PREDICTION OF COIL PERFORMANCE[a]

(*6 Row Plate Fin Coil*)

No.	D.B. In		W.B. In		D.B. Out		W.B. Out		T_r		Load-Btu/hr		lb Air/hr	
	Predicted	Actual	Predicted	Actual	Predicted	Actual	Predicted	Actual	Predicted	Actual	Predicted	Actual	Predicted	Actual
1	80.0	80.38	75.0	75.24	64.0	64.11	63.3	63.34	52.23	50.30	70,416	75,630	7260	7570
2	90.0	89.96	75.0	74.92	65.9	65.90	64.0	64.04	53.48	52.70	66,600	67,200	7260	7430
3	100.0	99.62	78.0	77.79	67.2	67.03	64.4	64.29	50.62	50.38	85,752	84,390	7260	7330
4	100.0	100.1	75.0	75.08	68.0	68.26	64.65	64.74	54.67	54.85	63,216	62,875	7260	7390
5	80.0	80.1	67.0	67.0	57.75	57.98	56.0	56.26	47.0	46.2	55,420	57,300	7260	7660

[a] Actual temperatures shown in the table are each the averaged result of 100 individual thermometer readings taken during equilibrium conditions at 5-min intervals for a period of 2 hours.

TABLE 5—PREDICTION OF COIL PERFORMANCE[a]

(*5 Row Spiral Fin Coil*)

No.	D.B. In		W.B. In		D.B. Out		W.B. Out		T_r		Load-Btu/hr		lb Air/hr	
	Predicted	Actual	Predicted	Actual	Predicted	Actual	Predicted	Actual	Predicted	Actual	Predicted	Actual	Predicted	Actual
1	90.0	90.30	75.5	75.57	69.3	69.55	65.2	65.23	52.5	52.5	69,826	70,660	7990	8090
2	80.0	80.56	73.0	72.77	64.6	65.35	62.7	62.71	49.8	48.97	65,882	66,100	7990	8165
3	105.0	105.30	76.7	76.67	72.4	72.84	64.5	64.50	48.0	48.04	82,923	80,930	7990	7905
4	100.0	100.19	78.2	78.23	72.8	72.89	66.7	66.67	50.9	50.42	81,504	81,310	7990	7970
5	100.0	100.18	78.0	78.43	75.9	75.95	69.6	69.60	57.9	56.70	61,540	64,620	7990	7910
6	85.0	85.04	71.0	71.03	64.8	65.18	61.0	61.01	48.5	48.17	61,230	60,830	7990	7955

[a] Actual temperatures shown in the table are each the averaged result of 100 individual thermometer readings taken during equilibrium conditions at 5-min intervals for a period of 2 hours.

TABLE 6—COMPARISON OF COIL PERFORMANCE[a]

(*3 Row Plate Fin Coil*)

No.	Entering Air Conditions		Air Flow lb/hr	Load Ratio S/T	Exit Conditions				Total Load Btu/hr		Refrigerant Temp. deg F	
					By Test		Calculated					
	D.B.	W.B.			D.B.	W.B.	D.B.	W.B.	By Test	Calc.	By Test	Calc.
1	103.25	85.29	7680	0.628	83.19	77.97	83.15	77.90	59,720	59,800	57.9	58.0
2	90.12	75.06	7750	0.832	74.52	69.99	74.35	69.92	35,320	35,300	52.9	57.6
3	90.31	76.86	7620	0.559	70.82	67.26	70.99	67.33	64,590	64,000	48.0	45.1
4	90.20	74.87	7950	0.918	75.58	70.46	74.80	70.20	32,890	35,100	58.9	58.2
5	84.22	73.15	4052	0.520	62.25	60.01	62.25	60.01	41,560	41,560	46.9	44.1
6	85.51	71.18	7800	0.647	64.36	60.68	65.10	61.12	61,940	59,900	35.7.	39 5
7	63.51	58.63	8020	0.575	53.99	52.29	53.60	52.02	29,780	30,900	44.5	39.8
8	90.32	75.39	5663	0.652	70.35	66.72	70.51	66.80	42,160	42,000	51.5	51.5

[a] Test temperatures shown in the table are each the averaged result of 100 individual thermometer readings taken during equilibrium conditions at 5-min intervals for a period of 2 hours.

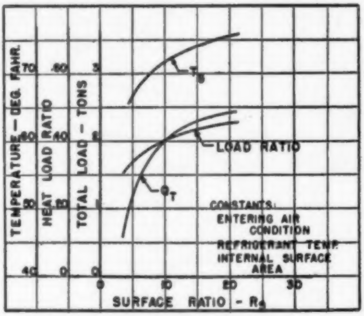


FIG. 6. EFFECT OF SURFACE RATIO ON COIL PERFORMANCE

ACCURACY OF HUMIDITY METHOD

If the equations of Table 1 are used for the solution of coil problems, air-side performance may be predicted well within the limits of accuracy of psychrometric determinations. Refrigerant-side performance (*i.e.*, refrigerant temperature) may be predicted within about two degrees suction temperature, which is within the tolerance of most commercially obtainable freon 12. Evidence of these statements is given in Tables 4, 5, and 6, which show a comparison of predicted coil performance vs. actual performance obtained by test. Table 4 shows results for a plate fin coil; Table 5, for a spiral fin coil; and Table 6, for a plate fin coil on which no rating tests were run. That is, the coil used to obtain the results of Table 6 was built according to certain specifications, and the performance was predicted simply from a knowledge of the entering air conditions, the air quantity, and the required load ratio.

The actual procedure by which the data of these tables were obtained is as follows: For Tables 4 and 5, coil rating tests were run to determine h_a and h_r.

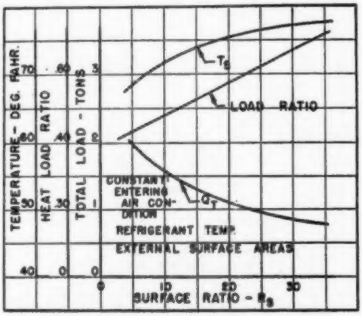


FIG. 7. EFFECT OF SURFACE RATIO ON COIL PERFORMANCE

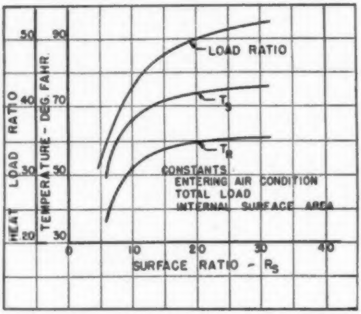


FIG. 8. EFFECT OF SURFACE RATIO ON COIL PERFORMANCE

Then through the use of these coefficients and the equations of Table 1, coil performance was calculated for certain specified conditions of entering air temperature, air quantity and refrigerant temperature. These conditions are indicated in the tables as *predicted values.* After these predictions had been made, tests were run to verify the calculated performance. The results of these tests are shown in the tables as *actual* values. In many cases the difference between the calculated and actual load can be traced directly to the inability to set the air quantity exactly at the required *predicted* value.

For Table 6 the procedure was somewhat different. No rating tests were run on this coil. But the coefficients had been previously determined for a similar coil design of different depth (6 rows) built by the same manufacturer. Therefore, to test the accuracy of the *Humidity Method,* especially in respect to effects of coil depth, the *calculated* values of Table 6 were determined. To obtain these *calculated* values, the entering air conditions, air quantity, and load ratio taken from tests on the three row coil were selected as *given* data, and

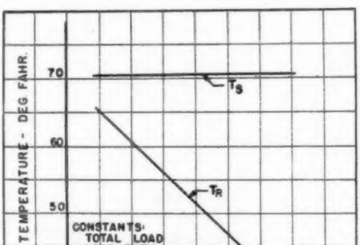


FIG. 9. EFFECT OF SURFACE RATIO ON COIL PERFORMANCE

exit air conditions, refrigerant temperature, and total load were calculated by use of the coefficients of the *six* row coil and the equations of Table 1. These *calculated* results were then compared with the actual test results as shown in Table 6.

It is evident from an examination of these tables that air-side accuracy is much more satisfactory than refrigerant-side accuracy. However, before refrigerant-side accuracy can be improved, much more research is needed on

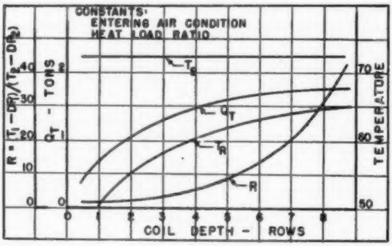


FIG. 10. EFFECT OF COIL DEPTH ON COIL PERFORMANCE

refrigerant coefficients and more control is required on refrigerant properties so that these properties will agree with published data.

ADAPTABILITY OF THE HUMIDITY METHOD

Although simplicity and accuracy are two important features of the *Humidity Method,* its general adaptability is no less an advantage. This adaptability permits the direct solution of many problems which are extremely difficult and tedious by other methods. The problem of high latent loads when there is no intersection of the load ratio line with the saturation curve is one such example which is worked out in the Appendix. In order to illustrate further the general application of the *Humidity Method,* the effect of various factors on the performance of dehumidifying coils has been plotted in Figs. 6 through 17. These curves are the results of many calculations made by K. C. Garman.[6] All the data for constructing these curves were obtained only through the use of the three equations given in Table 1. These curves do not represent test results and are not for any particular coil. They are simply based on calculations to demonstrate typical coil performance.

APPLICATION OF HUMIDITY METHOD TO DRY COOLING

Dry cooling problems are just as simply solved by the *Humidity Method* as by any other method. Since for dry cooling the initial and final dew-points

[6] Factors Affecting the Performance of Direct Expansion Coils in the Cooling and Dehumidification of Air, by K. C. Garman. (Thesis, Case School of Applied Science, 1944.)

are equal, the value of the fraction $(T_1 - DP_1) \div (T_2 - DP_2)$ loses its significance and should not be used. However, the value of R calculated from Equation 1 still applies, and the surface temperature and refrigerant temperatures may be calculated from Equations 2 and 3 in exactly the same manner as for cooling in which dehumidification takes place. The only limitation of the humidity method regarding dry cooling and dehumidification is that rating tests must be run with the coil dehumidifying. That is to say, the constants h_a and h_r must be determined from wet-coil tests. But once these constants have been determined for a particular coil design, they may be applied equally well to determine coil performance for either dry-cooling or dehumidification.

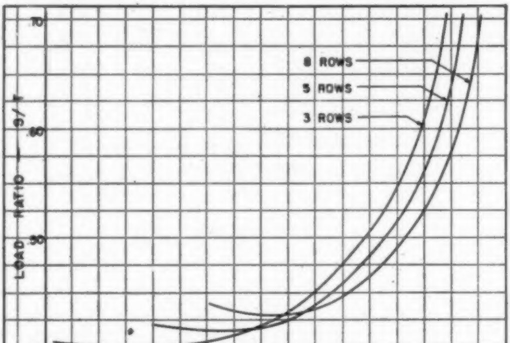


FIG. 11. EFFECT OF COIL DEPTH ON HEAT LOAD RATIO FOR CONSTANT ENTERING AIR CONDITION

CONCLUSIONS

Based on an examination of the curves given in Figs. 6 through 17 and upon other data given in the paper, the following general conclusions can be drawn:

1. Accurate coil rating and selection may be accomplished on the basis of three simple equations; and the constants (h_a and h_r) required for these equations may be obtained from a minimum number of simple coil tests. These constants depend only on fundamental coil design and need not be obtained for each different coil depth.

2. For direct expansion coils that may be required for either dehumidification or dry cooling alone, a surface area ratio between 15 and 20 is recommended because little effect on coil performance is obtained when the surface ratio is increased beyond 20. Figs. 6 through 9 show the effect of changing this area ratio from 5 to 30 by various means and for different conditions of coil operation. Near a value of 20, the refrigerant coefficient referred to air-side surface, $\frac{h_r}{R_a}$, becomes nearly equal to the air-side coefficient. Therefore little can be gained by a further increase of external surface because the refrigerant coefficient becomes the limiting factor in heat transfer.

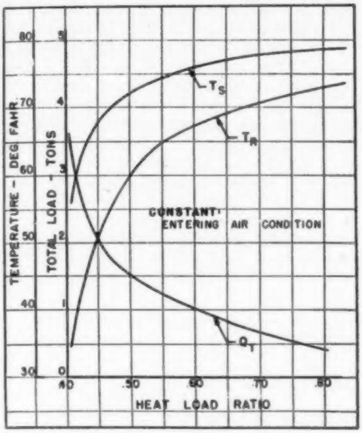
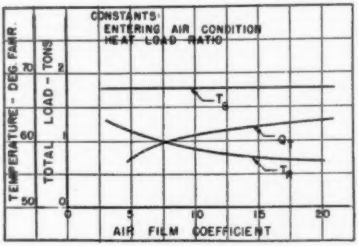


FIG. 12. EFFECT OF HEAT LOAD RATIO ON COIL PERFORMANCE

FIG. 13. EFFECT OF AIR FILM COEFFICIENT ON COIL PERFORMANCE



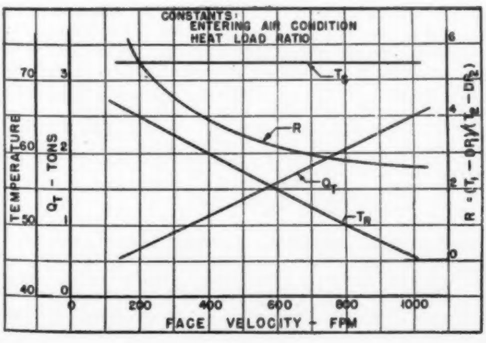


FIG. 14. EFFECT OF FACE VELOCITY ON COIL PERFORMANCE

3. There is little advantage gained in designing a coil more than 7 rows deep, if a surface area ratio of between 15 and 20 is used. This is shown in Fig. 10 where the load curve and refrigerant temperature curve level off rapidly after a depth of 7 rows is reached. This figure has been drawn for an average coil with ⅝ in. O.D. tubes, and therefore, does not necessarily apply to unusual designs which may have very closely spaced rows of smaller diameter tubes. For these special cases, the optimum coil depth should be chosen by constructing curves similar to those of Fig. 10 through the use of the equations of Table 1. For a general approximation which applies to all coils, however, the maximum coil depth may be selected on the basis of whatever depth is required to provide an air-side surface area of about 100 sq ft per square foot of face area.

4. For a given coil depth, there is a minimum sensible to total heat load ratio that may be obtained regardless of refrigerant temperature or load. This is shown in Fig.

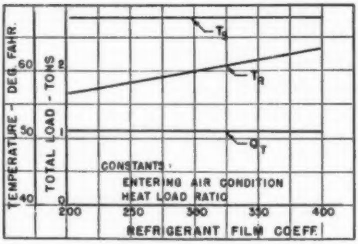


FIG. 15. EFFECT OF REFRIGERANT FILM COEFFICIENT ON COIL PERFORMANCE

11 which indicates that the minimum heat load ratio for a given type of coil surface increases with coil depth.

5. Steep load ratio lines require very low refrigerant temperatures as indicated in Fig. 12. Also, as just cited, shallow coils are usually required. This explains why reheat is often used in practical coil applications involving steep ratio lines. By using a deeper coil, a higher refrigerant temperature may be satisfactory and a smaller compressor may be used. Reheat may then be applied to the air leaving the coil to decrease the load ratio to the desired amount.

6. Increasing the value of the air film coefficient beyond about 15 Btu per hr per sq ft per deg F for a direct expansion coil having a surface area ratio between 15 and 20, has little effect on coil performance as indicated in Fig. 13. Since this coefficient increases with velocity, it might be reasoned that the air velocity is likewise limited at some value that would correspond to a coefficient of 15. However, an examination of Equation 1 and Fig. 14 will show that this is not the case. For a constant load ratio and constant entering condition, the coil load is practically a straight line function of the air velocity while the refrigerant temperature decreases in the same manner. The limiting face velocity for a coil may often be determined by the amount of carry-over of moisture that occurs at the downstream coil face. With usual coil designs, this velocity is about 600 fpm.

7. In spite of the fact that only limited data are available on refrigerant coefficients, little error will result in compressor selections based on these coefficients which have been determined from tests under typical operating conditions. This is because large changes in refrigerant coefficient have only a slight effect on refrigerant temperature. As shown in Fig. 15 a change of plus or minus 25 per cent in refrigerant coefficients

results in a change of only plus or minus 1.5 deg in refrigerant temperature. The properties of commercially obtainable freon 12 often vary by more than this amount.

APPENDIX

EXAMPLES OF COIL SELECTION PROBLEM

An industrial application requires the cooling of a certain quantity of air from a condition of 102 F dry-bulb and 85 F wet-bulb to a final condition of 80.5 F dry-bulb and 73 F wet-bulb. The air velocity across the coil is to be 400 fpm and coil data are as follows:

h_a = 10.7 at 400 fpm
h_r = 325

External surface area = 15 sq ft per sq ft of face per row of coil depth.
Ratio of external surface area to internal surface area = 15.

SOLUTION

1. Lay out the problem psychrometry as indicated in Fig. 16 and note that the minimum horizontal distance between the load ratio line and the saturation curve is 1.8 F dry-bulb at point A Fig. 16. This means that $T_2—DP_2$ in Equation 1 must not be less than 1.8. Therefore, Equation 1 should be solved for N to determine the proper number of rows to be used for the coil.

$$\frac{h_a \times A_s N}{0.243\, G_a} = \log_e \frac{T_1 - DP_1}{T_2 - DP_2}$$

$$= \log_e \frac{102 - 80}{1.8} = \log_e 12.22 = 2.5$$

Then substituting values for h_a, A_s and G_a, N may be found as follows:

$$\frac{10.7 \times 15N}{0.243 \times 1740} = 2.5 \text{ or } N = 6.58$$

2. This establishes the maximum whole number of coil rows that can be used as 6 and it is now possible to determine the actual location of the exit air conditions from Equation 1 by solving for the actual value of $T_2—DP_2$ for a 6 row coil.

$$\frac{10.7 \times 15 \times 6}{0.243 \times 1740} = \log_e \frac{102 - 80}{T_2 - DP_2} = 2.275$$

This establishes values of 9.78 for

$$\frac{T_1 - DP_1}{T_2 - DP_2} = R \text{ and } 2.25 \text{ for } T_2 - DP_2.$$

3. Next, the exit air condition at 57.3 F dry-bulb and 56 F wet-bulb as shown at B, is found by locating a point on the load ratio line at a horizontal distance of 2.25 dry-bulb degrees from the saturation curve.
4. The surface temperature may now be found from Equation 2.

$$T_s = \frac{9.78 \times 57.3 - 102}{8.78} = 52.3$$

5. The total coil load may be calculated from the enthalpy difference across the coil and the air quantity using the weight of dry air instead of the weight of the mixture.

$$\begin{aligned} Q_t &= G_{da}\,(h_1 - h_2) \\ &= 1700\,(49.24 - 23.77) \\ &= 43{,}200 \text{ Btu per hr per sq ft of face area.} \end{aligned}$$

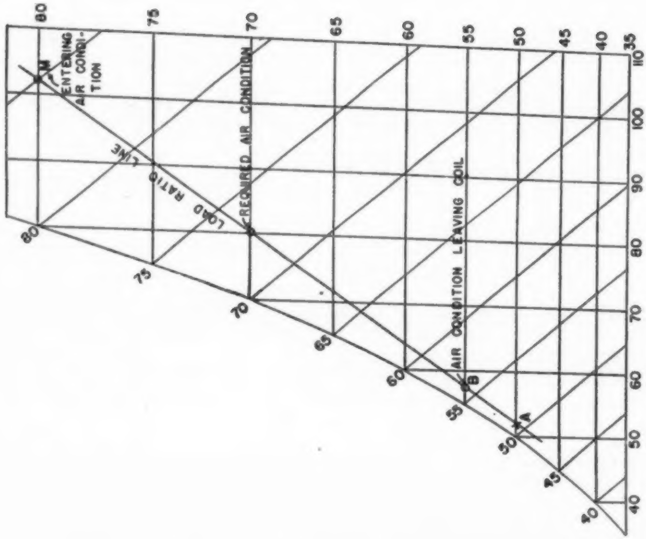


FIG. 16. PSYCHROMETRIC LAYOUT FOR COIL SELECTION

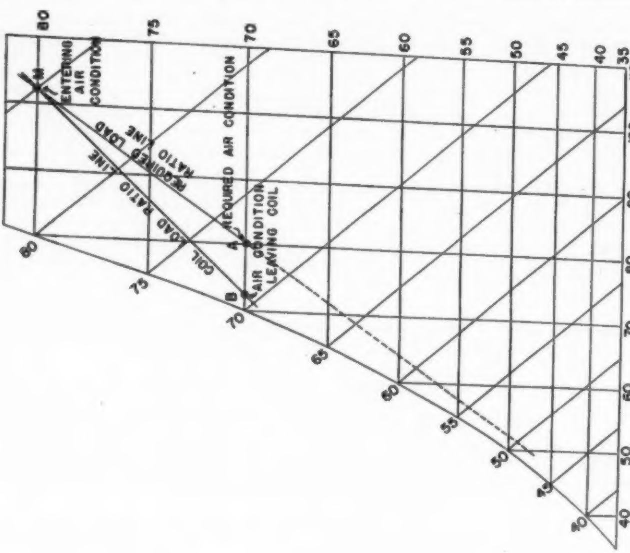


FIG. 17. PSYCHROMETRIC LAYOUT FOR COIL SELECTION

6. The refrigerant temperature may be found from Equation 3

$$\frac{43200}{15 \times 6 \times \frac{325}{15}} = T_s - T_r = 22.1$$

Therefore,

$$T_r = 52.3 - 22.1 = 30.2.$$

Thus a coil 6 rows deep operating at a refrigerant temperature of 30.2 F and a face velocity of 400 fpm is required and it will carry a total load of 43,200 Btu per hour per square foot of face area. The air conditions leaving the coil are too low for the conditions of the problem and therefore it is necessary to by-pass air at the entering condition to obtain the desired result of 80.5 F dry-bulb and 73 F wet-bulb.

Although the solution is satisfactory, it may be more desirable in some cases to use a higher refrigerant temperature and employ reheat to obtain the desired load ratio. Such a solution is shown in Fig. 17. In this case the coil load ratio line intersects the saturation curve and therefore a coil of any depth may be selected.

If a coil depth of 6 rows is maintained, the exit air conditions for the coil are indicated at point B Fig. 17 as 72.3 F dry-bulb and 70.8 F wet-bulb and the surface temperature will be:

$$T_s = \frac{9.78 \times 72.3 - 102}{8.78} = 69.0$$

The coil load will be:

$Q_t = 1700\ (49.24 - 34.66) = 24{,}800$ Btu per hour per square foot of face area and the refrigerant temperature will be found from Equation 3:

$$\frac{24800}{15 \times 6 \times \frac{325}{15}} = T_s - T_r = 12.7$$

Therefore,

$$T_r = 69.0 - 12.7 = 56.3.$$

Thus, for the case where reheat is used, a coil 6 rows deep operating at a refrigerant temperature of 56.3 F is required. The total coil load will be 24,800 Btu per hour per square foot of face area, but the actual effective load will be less by the amount of reheat required. Therefore, for a given load, a larger coil and more refrigerating capacity are required when reheat is used.

DISCUSSION

H. B. NOTTAGE, East Hartford, Conn. (WRITTEN): Within the published scope of this work, the author certainly seems to have come forth with a coil performance analysis method which is of utmost simplicity. But from the fundamental research interest, the unanswered questions and problems which come to attention through a critical study seem to offer unsolved problems of far-reaching importance to the Society. It is hoped that the author will assist by discussing a few points.

In dealing with *performance,* this paper omits a consideration of the pressure-drop problems, both in the air and refrigerant circuits of the coil. Should not the picture be completed in this respect? Such data should be of fundamental importance.

Also, in using the term *performance,* it would be helpful if various possible definitions of precisely what is meant thereby were set down. It seems that the various factors entering into performance might be variously combined or compared for different purposes, *i.e.,* is performance overall, air-side, or refrigerant-side, and how

would different judgments be offered in terms of air and refrigerant flow rates, heat transfer, pressure drops, fluid temperatures and pressures, surface areas, flow cross-sectional areas, tube geometry and arrangement, and metal temperatures? Is a test for particular desired conditions or construction always needed? How is the author's method related to design problems? Can performance be predicted?

The statement that coil performance analysis *involved only sensible heat coefficients* irrespective of simultaneous mass transfer and variations of wetted or frosted surface must have some fundamental basis which ought to be emphasized. If this is generally true, why is the humidity method not applied to cooling towers or evaporative condensers?

The equations of Table 1 imply that the coil surface temperature is a single and precisely-defined variable. It seems that considerable variations in surface temperature would be expected from point to point in most cases. What is the fundamental explanation of the apparent accounting for this?

In future work, it would be most helpful if the data were given fundamental interpretation in terms of the proper component heat-transfer conductances, air-side, metal, and refrigerant-side, and if such results were compared in basic terms to the available information thereon in the literature.

The relation of $h_a = CV^n$ is valid only for a constant air density. How did the test air density vary?

In the erratic variation of the refrigerant-side coefficient, what are the fundamental concepts or phenomena which would enter into a true explanation of this? May the method of analysis from Equation 3 of Table 1 not have some bearing on the erratic results? It seems that much further basic work is desirable on the refrigerant-side problem.

What limitations are there in the number of coil rows whose performance might be predicted by the subject method of this paper? Would Table 6 justify a general conclusion?

Does this method of analysis work with equal satisfaction for other basic types of transfer surfaces, such as flat plates or bare tubes?

Further, are there limitations on the range of air temperature, pressure or moisture content for which the humidity method will be accurate? Need this analysis be in any way tied down to a conventional psychrometric chart?

C. M. ASHLEY, Syracuse, N. Y.: This paper represents a contribution of much value to the art and the author is to be congratulated on the work which is presented.

One of the most interesting features of the paper is the apparent deviation of the test results as shown in Figs. 3 and 4 from those which would be expected using the Lewis equation, which would predict that the progressive state points would approach more nearly the saturation line. It has, of course, been recognized that the Lewis equation is only an approximation, but the magnitude of the deviation appears to be greater than can be justified by the order of approximation of the Lewis equation. In reading this paper it is well to keep in mind the fact that what has, in essence, been presented is an approximate method of analysis. The use of *film coefficients* is somewhat unfortunate since the *coefficients* shown do not strictly have physical significance as such. This is particularly true of the refrigerant-side coefficient in the evaluation of which there are at least three approximations of some importance. The air-side *film coefficient* represents a better approach to the physical reality, but still involves some approximation.

Of the various methods of analysis proposed, however, this appears to be one of the best and should well serve as a starting point for additional work. It has been my observation that almost all coil selection methods which, as this does, involve approximations, have certain limitations as to their area of use. It would, therefore, be most interesting if it could be determined that the present method of analysis could be extended with good accuracy to cover a wide range of temperatures, and to include the process of evaporative condensation, a wide range of coil depths,

various types of surfaces and surface ratios, and particularly to cover coils having a large variation of surface temperature.

Another reason why I place emphasis upon obtaining values as close to the true physical *film coefficients* as possible is that (1) this permits the correlation of data obtained from very diverse sources and (2) permits the prediction of performance of a whole series of coils from tests of a limited number of sizes; both of these points being of considerable commercial importance.

W. C. WHITTLESEY, Washington, D. C.: A discussion of this paper would not be complete without a brief resumé of the effect this method has had on naval design. Before the war, very little air conditioning was installed aboard naval vessels, and there was ample time in which to complete the few installations which were made. Equipment was specifically designed for each installation. However, wartime experience in the South Pacific demonstrated that a considerable increase in mechanical cooling applications was advisable. Equipment standardization and a simplified method of selection was necessary to accomplish this increase ôn our greatly expanding shipbuilding programs.

All the standard available methods of coil selection were investigated. Three prime factors appeared desirable: *First,* the use of effective temperature for the design condition rather than the generally accepted practice of specifying a fixed wet- and dry-bulb combination. *Second,* a constant refrigerant temperature; for example, if a single compressor plant were to serve a number of coils installed in separate spaces, each coil must be selected to operate at the same refrigerant temperature. The standard methods in general use proved unwieldy as they required considerable experience on the part of the designer or numerous trial calculations to select a coil for a given tonnage, heat ratio, entering air condition and refrigerant temperature. *Third,* standardization on a minimum number of units was vital. The procurement problem definitely eliminated the practice of specialized designs, and furthermore battle damage replacement equipment had to be provided at all repair facilities.

Case School of Applied Science was requested to develop a selection procedure that would fulfill the requirements, and the humidity method is the result. Its use enables the quick selection of a satisfactory cooling coil from a series of six coils. This series provides capacities up to about 20 tons with sensible heat ratios from 0.4 to unity.

The anxiety of the Navy Department to expedite the development of a simple and practical method for the use of design personnel with limited experience in heat transfer problems is believed to have been a major influence in discouraging refinements or allied investigations, that might normally be expected of basic research under more leisurely conditions.

G. L. TUVE, Cleveland, Ohio: The Society and Case School of Applied Science want to thank the Navy Department, Bureau of Ships, for their support of this project. The project was undertaken at the instigation of the Navy Department, and the Society research program has benefitted greatly from this association with the Bureau of Ships, Air Conditioning Section.

I should like to call attention to one very acute difficulty, from the viewpoint of the Committee on Research. As our research program increases in size and scope we have less and less time for adequate discussion of technical papers by people who are intensely interested in specific things. I should like to make an appeal to those of you who attend these meetings to submit your discussions in writing on this paper and on other papers. When you do not have an opportunity to present a discussion in the meeting, mail it to headquarters later. Unless we get the advantage of this discussion we are missing much of the value of our research papers. We are aiming to bring together the opinions of many Society members on each research subject. When written discussions are submitted they can be utilized by our technical ad-

visory committees and our Committee on Research and they are much appreciated by the authors themselves.

AUTHOR'S CLOSURE: There are a few critical points I would like to mention. One of these is the question about the approximation of Equation 3 in Table 1. I realize that the equation is approximate, but I am at a loss as to what to do about it at the moment because of inadequate data on film coefficients. I think one of the most important items is the mean temperature difference, as Mr. Ashley has pointed out. The surface temperature varies from row to row in the coil, while the refrigerant temperature may remain substantially constant. I have made many attempts to determine the mean temperature difference on this basis with no success. It seems to vary depending upon the load on the coil.

Regarding the application of the *Humidity Method* to problems involving depth, although it was not indicated in the paper, the method has been applied to depths from two to twelve rows and has worked satisfactorily. When a coil depth greater than seven rows is used, air leaves the coil so nearly saturated that for most practical purposes you may consider it represented by a point on the saturation curve. That is true with most coil designs. Of course, with special types of design, more than seven or eight rows may be required to bring leaving conditions near the saturation curve.

In answer to the question: "Could the method be applied to cooling towers?" I think something similar to it could be, if enough data were available on cooling tower surface areas. From what I have been able to determine about cooling towers, which I will admit is not very much, I have never been able to satisfy myself about the relative proportions of film or drop surface under various conditions of operation. If I could establish that in my mind, I think something might be done to extend the *Humidity Method* for application to cooling towers.

1272

No. 1272

A SURVEY OF TESTING METHODS AND RATING LIMITS FOR DOMESTIC HEATING DEVICES †

By R. S. Dill,* Washington, D. C.

THERE HAS been considerable progress in the formulation of testing and rating methods for heating devices during the past several years, much of which has been due to the interest of the various housing agencies of the Government. It is generally agreed that economy can be gained by installing heating devices or systems more nearly suitable in capacity for the houses they are to serve, than have been used in a large number of cases heretofore and interest in standards has increased because equipment and man-hours can be saved by reducing the number of kinds or sizes of heating devices manufactured. This is particularly important during the war period.

Introduction

A survey or summary such as this may serve a useful purpose in affording an over-all view of testing methods and rating limits, but it is not to be presumed that this paper constitutes a criticism of any device or test method or that the purpose is to show how one class of devices suffers in comparison with another. For instance, the fact that a coal stove may have a required efficiency lower than that of some gas-burning device is no reason why the stove should never be used. Gas is a highly refined fuel and high combustion efficiencies with it are to be expected, but in some regions gas is much more expensive per Btu than coal (or other fuels). For this reason, many people will continue to use coal stoves. These considerations indicate, if carried further, that a variety of equipment should be available, in order to permit flexibility in utilization, and standards for each type of equipment appear to be in order. These standards rightly have been based for the most part on performance. The objective is, and should be, to define the purpose of a device and leave it to the manufacturer or inventor to develop the best possible type.

Commercial Standards are based on a consensus of all interested parties including manufacturers, distributors, and users. The Trade Standards Division of the *National Bureau of Standards* is maintained to facilitate better understanding within the entire trade. More governmental influence than usual has been exerted toward standardization during the war period.

Testing methods and rating limits for a number of coal and oil burning domestic heating devices have been incorporated in Commercial Standards and Trade Standards, published by the *National Bureau of Standards,* within the last few years. These testing methods and rating limits have, therefore, re-

† Approved by *National Bureau of Standards Editorial Committee.*
* Chief, Heat Transfer Section, National Bureau of Standards. Member of A.S.H.V.E.

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TABLE 1. RATING LIMITS OF

DEVICE	SOURCE OR PUBLICATION	TEST FUEL	STACK Max Temp Fahr	STACK Min CO_2 %	MIN EFFICIENCY %	MAX DRAFT IN. WATER
Anthracite burner (domestic stoker)	CS 48–40[a]	Anthracite buckwheat No. 1 or No. 2	....	7.5	50	....
Oil burners (automatic mechanical draft)	CS 75–42	Heaviest approved by Underwriters' Laboratories	....	10.0	....	0.03 to 0.05 in comb'n chamber
Gas floor furnaces	CS 99–42	....	....	....	....	....
Space heaters (flue connected; with pot-type oil burners)	CS 101–43	No. 1 fuel oil (See CS 12–40 "Fuel Oils")	920 plus room	....	70 at 0.06 draft	0.08 in flue
Warm air furnaces with pot-type oil burners	CS (E) 104–43	Heaviest recommended by manufacturer				
Gravity circulation	....		....	10	70	....
Forced circulation	....		....	10	72	....
Forced air furnaces (solid fuel burning)	CS 109–44	Anthracite chestnut	830 plus room	....	55	0.06
Floor furnaces (pot-type oil burners)	CS 113–44	Heaviest recommended by manufacturer	780 plus room	10	70	0.02 to 0.06
Coal burning space heater (with or without jackets)	TS–3443[b]	Anthracite chestnut				
Surface fired	....	....	900	....	55	0.06
Magazine feed	....	....	900	....	50	0.06
Hot water supply boilers (hand fired)	TS–3560[b]	Anthracite chestnut or stove	....	....	....	....
Heating boilers (hand fired)	I=B=RCode[c]	Anthracite stove or size specified by manufacturer	....	....	58	....
Mechanically fired	I=B=RCode	No. 2 fuel oil or as specified for special burners	600	10	68	0.02 in fire box
Gas burning appliances	Approval requirements for central heating appliances[d]	Natural or manufactured gas or propane, butane or butane-air mixtures, in accord with design	480 plus room			
Steam boilers				....	75	....
Hot water boilers				....	75	....
Warm air furnaces Gravity	....			....	70	....
Forced	....			....	75	....
Floor furnaces			530 plus room			
Gravity	....	....		....	65	....
Forced	....	....		....	70	....

[a] CS = Commercial standard. Promulgated by *The National Bureau of Standards.* Available from the Government Printing Office, Washington D. C.

[b] TS = Trade Standards Division. (These proposed commercial standards in process of development.)

DOMESTIC HEATING DEVICES

SMOKE	SPECIAL REQUIREMENTS	TEST METHOD AND REMARKS
....	Ash loss limited to 7.5 per cent or less. Endurance test of 300 hrs required. Rating is 90 per cent of output under test. Piping and pick-up allowance: 25 per cent.	Test in insulated, round, cast-iron boiler for efficiency, etc. Pick-up in 60 min from banked fire is required.
....	Required CO_2 in flue gas: 10 per cent in laboratory, 8 per cent in field. Smoke test apparatus described in Underwriters' Laboratories, Inc., std., for domestic oil burners. (Subject 296.)	Test in suitable boiler. Draft allowed over fire: 0.03 in. water for capacities less than 5 gph. 0.05 in. water for capacities of 5 gph or more.
....	No test specified. Standard covers construction and installation. Compliance with American standard approval requirements for central heating gas appliances Z 21.13 is required. For sizing, a 10 per cent pick-up factor is applied.	
6% [e] *ICHAM*	Low fire fuel rate not more than 0.7 lb/hr or 25 per cent of high fire rate, whichever is greater. Minimum draft 0.02 in. water.	Output determined by indirect test. Smoke evaluated by method of *The Institute of Cooking and Heating Appliance Manufacturers.*
10% *ICHAM*	Maximum air temp rise gravity furnaces, 160 deg F. Allowable air temp rise for fan furnaces; 90 ± 10 deg F. Maximum draft; 0.06 in. water for natural draft furnaces or 0.04 in. water for forced draft furnaces; not less than 0.02 in. water for either.	Either direct or indirect test device — Flue gas temp deg F above room: Forced draft furnaces .... 780 Natural draft furnaces .... 300 to 800 Gravity .... 300 to 920
....	Attention interval: 8 hrs or more for surface fired, or 12 hrs or more for magazine feed furnaces. Outlet air temp; 70 to 100 deg F above room temp. Banking period: At least 12 hrs for surface fired and 24 hrs for magazine furnaces.	Output determined by direct test with bonnet pressure of 0.2 in. water without air filters. Heat exchanger temp limited to 830 deg F above room average and 930 deg F above room maximum.
6% *ICHAM*	Low fire fuel feed rate: 25 per cent or less of high fire rate. Flue gas temp: 300 to 780 deg F above room for high fire.	Output determined by indirect test. A 10 per cent pick-up factor is applied for installation.
....	For surface fired heater: Attention interval: 6 hrs, banking period: 12 hrs, for magazine heater: Attention interval 12 hrs, banking period, 24 hrs. Heat exchanger temp limited to 1000 F or lower than red heat.	Output determined by indirect test. Jacket temp limited to 400 F except near door frame or flue collar.
....	Draft adjusted to simulate chimney action according to a curve. Rating is based on 6-hr firing period. Ratings are expressed in gallons of water per hour heated 100 deg F.	Direct test. Discharged water at 150 ± 20 F. Boilers up to 75 gpm capacity are mounted on weighing scale for test to facilitate observation of combustion rate.
....	Draft adjusted to simulate chimney action for observed flue gas temps. Attention interval: 4 to 7.5 hrs; piping and pick-up factor: 1.60 to 2.36. These factors vary inversely as boiler size.	Steam is condensed and weighed for boilers of less than 1,000,000 Btu per hr of capacity. For larger boilers, feed water may be weighed. This applies to mechanically fired boilers also.
....	Piping and pick-up factor: 1.34 to 1.56. Factor varies inversely with boiler size.	Test with high pressure atomizing burner unless designed for another type.
....	Test at atmospheric pressure. Required steam quality: 97 per cent.	Direct test. Feed water weighed.
....	Test with discharged water at 180 ± 2 F.	Direct test. Calorimetric method.
....	Air temp rise, without automatic cut off: 160 deg F. Automatic cut off, if used, must function at 250 F.	Direct test. Air delivery and temp rise are measured.
....	Discharge air temp: 70 to 100 deg ± 2 deg F above room temp.	Direct test, as above. Heating element temp limited to 875 F for either gravity or forced furnaces.
....	Air temp limited to 350 deg F above room temp one inch above grille.	Direct test, as above.

[c] I=B=R = Institute of Boiler and Radiator Manufacturers.
[d] Requirements approved by the *American Standards Association*, sponsored by *The American Gas Association*.
[e] *ICHAM = Institute of Cooking and Heating Appliance Manufacturers.*

ceived consideration by representatives of the interested industries, as well as by representatives of consumers including the Government agencies. This is not to say that these testing methods and rating limits have received universal approval, but at least they have received considerable support among the groups mentioned.

Some branches of the industry have established and have maintained their own standards for a long time, so that some items are covered by industry standards and not by Commercial Standards promulgated by the *National Bureau of Standards.* Chief among these are heating boilers and gas-burning devices in general. Probably the testing methods and rating limits contained in the code of the *Institute of Boiler and Radiator Manufacturers* are more widely used at the present time than any others for heating boilers, except gas-burning boilers. Gas-burning boilers and other heating devices are covered by the Approved Requirements for Central Heating Gas Appliances, sponsored by the *American Gas Association,* and approved by the *American Standards Association.*

Tests of devices for compliance with Commercial Standards (published by the *National Bureau of Standards*) are conducted in laboratories either maintained or chosen by the manufacturers. Very often an effort is made to simplify the specified testing procedure in order to minimize the trouble and expense to manufacturers who wish to conduct their own tests.

The rating limits for various classes of equipment have been tabulated in Table 1. As some of the figures given may not be comparable with each other, on account of differences in test methods, a more complete statement of the test methods and requirements for various devices is given as follows:

DOMESTIC BURNERS FOR PENNSYLVANIA ANTHRACITE (UNDERFED TYPE)

*Commercial Standard—CS*48-40: According to this Commercial Standard, anthracite stokers are installed for test in a round, sectional, cast-iron boiler, having three intermediate sections, insulated with 1½ in. of asbestos or equivalent. The stoker is tested at 50 per cent or more of boiler capacity at some fuel rate exceeding 50 per cent of its maximum for some period exceeding 4 hours. Under this condition, the over-all efficiency of the stoker and boiler is required to exceed 50 per cent, the ash loss is limited to 7½ per cent and the carbon dioxide content of the flue gas must be 7½ per cent, or more, by volume. The combustion rate is required to be at least 13 lb per hour per square foot of horizontal projected area of the firepot, including the ash ring, and such operation must be maintained at least nine hours. The test is made with anthracite No. 1 buckwheat or No. 2 buckwheat unless the stoker is especially designed for other fuels. The electric energy consumption is limited to 18 kw hours per short ton of coal burned when the stoker is operated at 59 per cent or more of maximum rating. Controls required are such that the stoker will maintain a fire indefinitely and that, when the stoker resumes operation after a 12-hour banking period, the time required for the stack temperature to reach normal shall not exceed 60 min.

AUTOMATIC MECHANICAL-DRAFT OIL BURNERS DESIGNED FOR DOMESTIC INSTALLATIONS

*Commercial Standard CS*75-42: According to this Commercial Standard, oil burners of the type mentioned are required to be capable of producing 10 per cent CO_2 in the flue gas during a laboratory test and 8 per cent CO_2 in the flue gas in field installations.

Oil burner testing apparatus is maintained by the Underwriters' Laboratories, Inc., Chicago, which includes means for testing these devices for radio interferences and

for smoke production. The smoke testing apparatus is described in Underwriters' Laboratories, Inc., Standard for Domestic Oil Burners (Subject 296).

For testing, the burner is mounted in a suitable boiler and is operated on the heaviest oil recommended for it by the Underwriters' Laboratories, Inc., the draft must not exceed 0.03 in. water for burners tested at 5 gal of oil per hour or less, nor exceed 0.05 in. water for burners tested at more than 5 gal of oil per hour.

FLUE-CONNECTED OIL BURNING SPACE HEATERS EQUIPPED WITH VAPORIZING POT-TYPE BURNERS

*Commercial Standard CS*101-43: According to the provisions of this Commercial Standard, heaters of the type covered are set up for test in a standard Underwriters' booth or so-called *black-corner.* The efficiency and output are based on flue gas analysis and temperature and fuel burning rate. The heater is required to develop an efficiency of 70 per cent or more when operated with 0.06 in. of draft or at a draft recommended by its manufacturer except that the draft must not be less than 0.02 in. or more than 0.08 in. water during the test. Smoke produced is required to be 6 per cent or less by the *Institute of Cooking and Heating Appliance Manufacturers'* test. It is required that unburned fuel gases shall not occur in the flue products in sufficient quantities to be measurable by recognized methods as unburned gas or vapor in excess of 0.2 per cent by volume or to result in failure of the observed CO_2 and O_2 values to check by more than 0.3 per cent. Stack temperature during the high fire test is limited to 920 F above room temperature. The test fuel is not heavier than Commercial Grade No. 1 oil (*Commercial Standard CS*12-40). *High fire* is an oil burning rate at 0.06 in. draft or at a draft recommended for high fire by the manufacturer. *Low fire* is defined for manually operated heaters as a burning rate of 0.7 lb of fuel per hour or less or not in excess of 25 per cent of the high fire rate, whichever is greater. Thermostatically operated heaters on pilot fire and manually operated heaters on low fire are required to meet the requirement for smoke specified for high fire.

This Commercial Standard contains a field test table, to be used as a basis for judging the acceptability of the performance of installed heaters, based on flue gas temperature and CO_2 content.

Also incorporated is a table of altitude correction factors for use in predicting the capacities of heaters at various altitudes up to 7000 ft above sea level.

WARM AIR FURNACES EQUIPPED WITH VAPORIZING POT-TYPE OIL BURNERS

*Commercial Standard (Emergency) CS(E)*104-43: This Commercial Standard covers both gravity flow and forced circulation furnaces with either natural or mechanical draft.

A forced circulation furnace is set up for test in accord with its manufacturers instructions, with air filter and humidifier in place, if the design includes them. The humidifier, if present, is left dry for the test. The furnace is then equipped with testing equipment including a discharge duct with a damper and, during the test, the bonnet pressure is regulated in accord with the following table.

AIR DELIVERY (Cfm)	EXTERNAL STATIC PRESSURE (IN. WATER)
0 to 800	0.12
Over 800 to 1600	0.20
Over 1600 to 3000	0.24
Over 3000 to 6000	0.30

The fuel-burning rate is adjusted to give an air temperature rise of 90 ± 10 F.

At maximum output the draft is required not to exceed 0.04 in. water for mechanical draft furnaces or 0.06 in. water for natural draft furnaces and to be not less than 0.02 in. water for either. The flue gas temperature must be between 300 F and 880 F above room temperature for natural draft burners and not more than 780 F above room temperature for mechanical draft burners. The CO_2 in the flue gas must be at least 10 per cent and smoke is limited to 10 per cent by the *I.C.H.A.M.* test.

The efficiency and output may be determined by either the direct or the indirect method. When the direct method is used, either the test specified in *American Society*

of Refrigerating Engineers' Circular No. 16, (Standard Methods of Testing and Rating Self-Contained Air Conditioning Units for Comfort Cooling) or another wind tunnel method, approved by the manufacturers' representatives on the standing committee, may be used. When the indirect method is used, the observed efficiency and output are termed gross efficiency and gross output. The gross efficiency is required to be at least 70 per cent for gravity furnaces and 72 per cent for forced-air furnaces. Bonnet capacity is found by deducting 5 per cent from the gross output.

Gravity furnaces are set up for test and fitted with 8, 10, or 12 in. leader pipe collars having an aggregate area not greater than

$$\frac{0.7 \times \text{Btu input}}{111}$$

The figure 111 is intended to represent the capacity of each square inch of leader pipe, in Btu per hour, for first-story gravity flow connections.[1] The air temperature rise is limited to 160 F above room temperature at high fire.

The draft is maintained between 0.02 and 0.06 in. water and the flue gas temperature, between 300 and 920 F, inclusive, above room temperature. The CO_2 in the flue gas is required to be at least 10 per cent and the smoke is limited to 10 per cent by the *I.C.H.A.M.* test.

SOLID-FUEL-BURNING FORCED-AIR FURNACES

*Commercial Standard CS*109-44: Under this Commercial Standard, furnaces of the type covered are set up and tested with chestnut anthracite and without air filters. Each furnace is tested for maximum rating and for banking. The minimum rating is not established by test but is computed on the assumption of a fuel-burning rate 3 times that observed during the banking test and an efficiency equal to that observed during the maximum capacity test. A separate test is required to find whether or not any point on the air filter exceeds 90 F above room temperature when the furnace is operated at maximum capacity.

The maximum capacity is determined by the direct method; the air flow and temperature rise through the furnace being measured and the heat output computed from these observations. The test consists of one preliminary and three firing cycles. The end of a cycle is defined as that instant when the difference between the stack temperature and the room temperature drops to 75 per cent of the difference between the flue gas test temperature and room temperature. The flue gas test temperature is determined during the preliminary cycle and is defined as the maximum temperature at which the furnace complies with the rating limits described below.

The efficiency is required to be 55 per cent or more when the draft is not more than 0.06 in. water and when the stack temperature does not exceed 830 F above laboratory temperature. The bonnet pressure is set at 0.2 in. water and the air temperature rise must be between 70 and 100 F. The fireproof or other metal heat exchanger parts must not exceed 930 F above room, maximum, or 830 F above room on the average. The jacket temperature is limited to 230 F above room temperature except at points above the firing door and within 6 in. of the sides of the firing door frame and flue collar. During the banking test, the heat output is required to be not more than 25 per cent of that observed during the maximum output test, based on fuel consumption. The required banking periods are 12 hours for surface fired furnaces and 24 hours for magazine feed furnaces.

OIL-BURNING FLOOR FURNACES EQUIPPED WITH VAPORIZING POT-TYPE BURNERS

*Commercial Standard CS*113-44: This applies to flue connected floor furnaces either with or without mechanical draft, forced circulation, or automatic controls.

Such furnaces are required to operate on high fire with an efficiency of not less than 70 per cent with a draft not exceeding 0.06 in. water, or less than 0.02 in. water and on low fire with a fuel rate not exceeding 25 per cent of that observed on high fire. Smoke is limited to 6 per cent by the *I.C.H.A.M.* test, and the efficiency is determined by the indirect method.

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[1] HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1944, p. 360.

The Commercial Standard contains also a table of factors for correlating fuel-burning rates with altitudes above sea level.

PROPOSED COMMERCIAL STANDARD (EMERGENCY) FOR COAL-BURNING SPACE HEATERS

TS-3443: This proposed standard covers space heaters with capacities of 60,000 Btu per hour or less. Tests are made with chestnut anthracite and the required efficiency is 50 per cent or more for magazine heaters and 55 per cent or more for surface fired heaters. Flue gas temperature is limited to 900 F and the draft to 0.06 in. water and metal to temperatures below 1,000 F or below red heat and the jacket, if used, to 400 F or less except at points within 6 in. of the firing door, door frame, top grille or flue pipe. Attention periods of at least 6 hours for surface fired and of at least 12 hours for magazine heaters are required. Surface fired heaters must bank for at least 12 hours and magazine fed heaters for at least 24 hours with a combustion rate of not more than 25 per cent of that required for maximum rating. The minimum rating is computed with the assumption of a fuel-burning rate three times that observed during the banking rate and an efficiency equal to that observed during the test at maximum rating.

RECOMMENDED COMMERCIAL STANDARD FOR TESTING AND RATING HAND-FIRED HOT WATER BOILERS

TS-3560*a*: This proposed standard covers solid-fuel-burning hot water boilers with capacities not exceeding 450 gal of water per hour with a temperature rise of 100 F.

Maximum output is defined as the average number of gallons of water which the device can heat through a temperature range of 100 F in one hour with continuous draw off of water, with the ashpit door closed and the ashpit damper open.

Minimum output is defined as the average number of gallons of water per hour heated 100 F with continuous draw off and with the fire banked, during a 12-hour period.

The tests are made with charcoal kindling and with anthracite chestnut or stove coal, as recommended by the manufacturer. Heaters with capacities less than 75 gal per hour are mounted on scales for test in order to facilitate measurement of the fuel-burning rate.

At least two tests are required, each including a preliminary cycle and a test cycle. During a test, the discharged water temperature is regulated, by adjusting the flow, to 150 F $\pm$20 F. Correction for rating in gallons per hour at 100 F rise is then made by a formula based on direct proportion. Each test cycle begins when coal is shovelled on top of 3 to 6 in. of hot fuel (or 25 to 35 per cent of test fuel charge) remaining on the grate from the preliminary cycle. Adjustment of charge and/or draft is made so that test cycles run for 6 hours plus or minus 15 min. For boilers with capacities less than 75 gal per hour of capacity, the test cycles are considered to have ended when the charge of test fuel has been consumed. This is indicated by a loss of weight by the boiler and the fuel it contains equal to that of the test charge minus the ash it contains, estimated from an analysis of the fuel.

For boilers of 75 gal or more per hour of capacity, test cycles are considered ended when the thickness of the fuel bed, after shaking the grates, is equal to the thickness of the kindling bed from the preliminary cycle, plus or minus 25 per cent.

I=B=R TESTING AND RATING CODE FOR LOW PRESSURE HEATING BOILERS

These codes are available from the *Institute of Boiler and Radiator Manufacturers,* New York 17, N. Y. The testing and rating code covers heating boilers of all sizes, both steam and hot water, hand-stoked, and oil fired.

Ratings of hand-fired boilers are based on a drive test with an assumed chimney height. The draft, usually obtained by means of a fan or blower, is regulated to simulate the action of a chimney of the assumed height by means of curves given in the code. The test is made with anthracite. An efficiency of 58 per cent is required. The firing period, like the assumed chimney height, is different for boilers of different sizes and is indicated by curves contained in the code. Boilers with a gross output of less than one million Btu per hour are tested by condensing and weighing the steam. Large boilers may be tested by weighing the feed water.

Oil-fired boilers are tested with No. 2 fuel oil with CO_2 in the stack gases set at 10 ± 0.2 per cent and with a flue gas temperature not exceeding 600 F. An efficiency

of 68 per cent is required. The rating for a boiler when stoker-fired is assumed to be the same as that obtained for it by test with oil fuel.

The test yield *Gross I=B=R outputs.* The gross I=B=R output is in each case converted into a net I=B=R rating by means of curves given in the code. The net I=B=R rating is less than the gross I=B=R output by a factor intended to account for the piping loss and pick-up.

SIMPLIFIED PRACTICE RECOMMENDATION R157-37

Steel heating boilers are usually rated by the formula contained in the *Simplified Practice Recommendation R*157-37, entitled *Steel Horizontal Firebox Heating Boilers.* The rating of a hand-fired boiler in square feet EDR is defined as 14 times the heating surface in square feet, and that of a mechanically-fired boiler as 17 times the heating surface, in square feet. Compliance with *A.S.M.E.* safety codes is required.

AMERICAN STANDARD APPROVAL REQUIREMENTS FOR CENTRAL-HEATING GAS APPLIANCES

This code or list of requirements is approved by the *American Standards Association* and sponsored by the *American Gas Association,* New York 17, N. Y. It contains construction and performance requirements for gas-burning boilers, warm air furnaces, and floor furnaces.

Codes are contained in the requirements for testing these devices with natural gas (1135 Btu per cu ft), with manufactured gas (535 Btu per cu ft), with butane or propane, and with a butane-air mixture. Test pressures are specified for each of the test gases indicated. The efficiencies and capacities of devices tested are directly measured, and efficiency is based on the higher heating value of the fuel gas. Carbon monoxide in the flue gases is required to be less than 0.04 per cent of an air free sample.

For steam boilers, the feed water is weighed and the weight used per hour is multiplied by the difference in enthalpy between the steam and feed water to determine the output. Hot water boilers are tested by the calorimetric method; the water flow rate is metered and its temperature rise is observed. The air flow rate for warm air furnaces, whether gravity or forced flow, and for floor furnaces is measured and its temperature rise is observed.

The efficiencies required for these appliances are as follows:

APPLIANCES	EFFICIENCY REQUIRED, PER CENT
Steam Boilers	75
Hot Water Boilers	75
Warm Air Furnaces	
Gravity	70
Forced	75
Floor Furnaces	
Gravity	65
Forced	70

Boilers may be either steel or cast-iron. Steam boilers are tested at atmospheric pressure and the steam quality is required to be at least 97 per cent. Each test is of 3 hours duration. Feed water is supplied from a tank, on a weighing scale, through an automatic water feeder. A complete heat balance is made but the rating is based on the steam produced. There is no allowance in the rating for piping and pick-up. The square foot EDR is defined as 240 Btu per hour. The flue gas temperature is limited to 480 F above room temperature.

Hot water boilers are tested with the discharged water at 180±2 F. Two tests are required, each of one hour duration. A complete heat balance is made but the rating and efficiency are based on the water flow rate and temperature rise. There is no piping and pick-up allowance included in the rating. The square foot of hot water radiation is defined as 150 Btu per hour. The flue gas temperature is limited to 480 F above room temperature.

Gas-burning forced air or fan-type furnaces are tested with air filters, and other accessories included in the design, in place. For test, a furnace is equipped with one or more outlet ducts in which thermocouples are installed to measure the discharge

air temperature. Restrictions are applied so that the furnace operates against a bonnet pressure of 0.2 in. water. A warm up period of one hour and two tests, each of at least one-half hour duration are required. The air temperature during the test is maintained between 70 and 100 F above room temperature and the air volume is measured by means of a meter in the inlet duct. Air filter face velocities are limited to 300 fpm or less. Fan furnaces are required to be equipped with automatic devices which will shut off the gas supply to the main burner at some outlet air temperature not in excess of 250 F and to turn on the gas when the air temperature drops to 135 F. The flue gas temperature is limited to 480 F above room temperature for either gravity or forced circulation furnaces. Heat exchanger metal temperatures are limited to 875 F and are required to reach temperatures of 178 F or more, in from 15 to 20 min from a cold start, depending on the inlet air temperature.

Gas-burning gravity circulating furnaces are set up for test with warm air ducts with an aggregate area not exceeding that obtained by means of the following formula.

$$A = \frac{0.75q}{111}$$

A = total area of warm air ducts, square inches.
0.75 = assumed efficiency.
q = manufacturer's input rating, Btu per hour.
111 = *National Warm Air Heating Association's* recommended heat carrying capacity for first-floor leaders, Btu per square inch.

The openings are made to accommodate 10 in. pipe except that one opening may be smaller so that the required area can be approximately obtained. Sufficient pipe is attached to each opening to obtain an effective rise of 24 in. above the center of the opening. Outlet air temperatures are obtained by means of five thermocouples in each duct. Tests are of one-half hour duration, preceded by either a one hour warm-up period or 15 min between tests.

The furnace is operated first with gravity air flow and the observed air temperature rise is duplicated with an air metering device connected to the air inlet. The discharged air temperature is limited to 160 F above room temperature unless the furnace is equipped with an automatic device which will shut off the main burners when the outlet air reaches a temperature not in excess of 250 F.

Gravity flow gas floor furnaces are set up for test with a collar 2 in. high around the warm air discharge opening, with nine thermocouples arranged in equal areas inside the collar one inch above the grille. The floor furnace is then operated and the temperature rise and heat input are observed. The collar is then replaced by a stub duct, 28 in. high, containing thermocouples, one for each 16 sq in. of cross-sectional area, in a plane 24 in. above the grille. The 9 thermocouples, 1 in. from the grille, are replaced and an air metering device is connected to the cold air inlet. The operation previously observed is then duplicated and the efficiency and output are computed, based on the average air temperature 24 in. above the grille and on the rate of air flow indicated by the meter. Tests are of one hour duration, with an equal warming-up period preceding the first test and a 15 min warming-up period preceding subsequent tests.

Fan-type gas floor furnaces are set up for test with a duct 28 in. high connected to the warm air outlet and air temperatures are measured with 16 or more thermocouples, none more than 6 in. from another, in a plane 24 in. above the grille. Air velocities in the test duct are determined by means of a velometer or pitot tube and temperatures in the duct are weighted for velocity. Ratings are based on air flow, determined with an air meter in the inlet duct, and the observed temperature rise. The air temperature is maintained between 70 and 100 F ± 2 F above room temperature during tests. Filters and humidifiers, if used, are in place during tests. Filter air velocity is limited to 300 fpm or less.

Floor furnace outlet air temperature is limited to 350 F above room temperature, measured one inch above the grille as described previously. Fan furnaces without automatic shut-offs must meet this requirement with the fan stopped. Flue gas temperature for floor furnaces is limited to 530 F above room temperature.

None of the foregoing outlines is intended to contain all of the requirements and information to be found in the publication to which it relates. The pur-

pose is to give those interested a broad view of how heating devices are tested and rated for heating capacity. Most of the publications discussed also contain requirements concerning construction, ease of operation, and safety. Those contemplating test work should, of course, obtain copies of the code, and requirements or standards applicable to the particular devices to be tested.

DISCUSSION

J. E. AXEMAN, Williamsport, Pa. (WRITTEN): The author has made a very able summary of the testing methods and rating limits for domestic heating devices and this paper should provide a ready reference for those interested in these codes.

It is evident from the author's summary that considerable progress in establishing codes has been made in recent years and this work has certainly been a great contribution to the domestic heating equipment field where various and sometimes questionable ratings were prevalent.

The extension of these codes to other equipment should be encouraged as they afford a necessary protection to all parties concerned in the purchase of equipment. This protection is particularly desirable in the domestic heating field where, unfortunately, purchase and installation of the equipment is at times made without the services of a specifying engineer or competent heating contractor; and even when the purchase is made by competent parties there should be some recognized standard to determine true output of equipment.

While it is not the purpose of the author's paper to discuss the technical aspects of these codes, it is significant to note that relatively low efficiencies appear to be acceptable. It is to be hoped, however, that this will be an inspiration to the industry to develop equipment showing higher efficiencies rather than equipment designed just to meet the minimum efficiencies acceptable under these codes.

M. W. MCRAE, Chicago, Ill. (WRITTEN): The author was modest in his statement that, "A summary such as this may serve a useful purpose in affording an over-all view of testing methods and rating limits, . . ." Unquestionably, this compilation will be very useful to those of us engaged in the testing and rating of domestic heating equipment and also to others in related fields.

As the author points out, it is obvious that a survey of this type should not be construed as criticism of any one test method; neither should it be interpreted as a comparison of one type of fuel or equipment with another. However, the tabulation in Table 1 brings out quite forcefully the wide variation in rating limits for heating devices having the same general purpose and fired with the same type of fuel. For example, the minimum efficiency permitted for coal-burning space heaters and warm air furnaces is 50 or 55 per cent, depending on whether the unit is magazine fed or surface fired, whereas, for steam and hot water boilers the minimum is 58 per cent.

At first glance, this difference does not seem to be significant, but when it is remembered that the efficiency curve is relatively flat, it becomes evident that a variation of 3 per cent in efficiency may change the allowable rating as much as 20 per cent.

Similarly, the maximum allowable stack temperature is much higher for the oil-fired, warm air furnaces than for steam and water boilers using the same fuel. The maximum stack temperature permitted for these warm air units varies from 780 to 920 F plus room temperature. This latter figure means an actual stack temperature of approximately 1000 F and, at a minimum efficiency of 70 per cent as specified in Commercial Standard 101-43, and disregarding all other losses, the CO_2 for these heaters would need to be at least 12½ per cent as indicated by the combustion curves shown in a circular [2] published by the University of Illinois. In comparison, the maximum stack temperature permitted for steam and hot water boilers, when oil-fired, is

[2] Combustion Efficiencies as Related to Performance of Domestic Heating Plants, by Kratz, Konzo and Thompson. (Circular Series No. 44, University of Illinois Engineering Experiment Station, p. 10.)

600 deg including the room temperature. This is quite closely in accord with the 480 and 530 deg plus room temperature permitted by the *American Gas Association* for central heating appliances.

In summarizing these comments, there are three points that should be emphasized as follows:

1. An extremely wide range of maximum allowable stack temperature is permitted by the various testing and rating methods currently in use.
2. The limitations imposed on their own equipment by manufacturers' groups, such as the *Institute of Boiler and Radiator Manufacturers* and the *American Gas Association*, appear to be much more conservative than similar requirements listed in Commercial Standards.
3. To insure equivalent performance of domestic heating devices utilizing the same fuel, it would seem desirable to establish these limits on a more uniform basis.

I would like to repeat that this survey is of great importance and we are indebted to the author for tabulating the available information so that these discrepancies are so obvious.

R. K. THULMAN, Washington, D. C. (WRITTEN): The author has prepared a very able and simple recapitulation of the testing and rating limits for domestic heating apparatus. A summary of this kind, in addition to affording an over-all picture of the domestic heating field, serves two additional useful purposes. *First,* it emphasizes the need for coordination of the existing codes and standards. *Second,* it points out the need for codes and standards for apparatus presently not covered.

The survey covers apparatus used principally in residences and mostly in small residences. In the field of small home construction the buyer is less protected by adequate specifications and architectural and engineering advice than is the case in the more costly field and in the commercial field. The home buyer rightfully expects that the heating system in the house which he purchases will be safe, efficient, economical, and convenient of operation and will provide heat throughout the dwelling in an acceptably uniform manner.

The service of the heating industry ultimately expresses itself to the consumer in these terms. The codes and standards covered by the survey are tangible expressions and a voluntary attempt to meet a recognized obligation by the industry to the public. These codes and standards are a recognition of a responsibility which goes farther than that which attaches to the manufacture and sale of any one of the several devices which may collectively supply the heating service. In other words, the formulation of these separate codes, without reference one to the other, threatens to perpetuate a division or even a lack of responsibility which has characterized the industry in the past. Several devices may be actually needed to constitute a complete heating system. The need for coordination appears clearly in any such case and suggests the necessity for certain collective or comprehensive standards that will blend the separate individual codes into an effective and proper whole.

Both the coordination of existing codes for this purpose and the initiative in the preparation of codes to fill in the gaps should, in my opinion, be an obligation of the A.S.H.V.E. The Society alone seems to be the logical agency to coordinate the wants and needs of the industry with the wants and needs of home owners and yet for some reason completely unclear to me, the Society has avoided this particular opportunity for public service.

Both the Society and the industry face a task of public education, it is true, but the public has for some time been trying somewhat inarticulately to educate the industry. The heating industry complains that the percentage of over-all cost budgeted for heating is too small to enable it to furnish the full comfort which it is capable of providing. One of the chief causes of complaint from the home owner on the other hand is the failure of his heating system to perform as he had been led to believe it should.

The situation seems to be one of simple cause and effect and I recognize that it can be argued which is the cause and which is the effect.

The over-all view which the author's paper affords suggests that this situation

could largely be overcome and that final results beneficial both to the public and heating industry would result if this Society acts as the coordinating agency to reconcile the inconsistencies in existing codes and standards, to initiate the preparation of codes and standards now lacking, and to combine the results in a residential heating standard.

R. C. CROSS, Chicago, Ill.: This condensed compilation of testing codes is useful information, particularly to those engaged in the testing of numerous lines of domestic heating devices, and Mr. Dill is to be highly commended for his efforts.

In viewing the basic concept of the establishment of test codes and standards it is necessary to consider their definitions. Although commonly used synonymously, closely related as to purpose, and at times combined in common text, a code may be defined as a body of classified laws, whereas, a standard is defined as an established rule.

A code of classified laws by which a device is tested for its functional performance should be: (1) Consistent with actual conditions of usage; (2) Practical in application to permit the average manufacturer or laboratory to utilize it.

A standard, or established rule, is a yardstick by which a device may be evaluated as to its quality of performance or construction. It provides a direct means of comparison for intelligent selection of merchandise. It should incorporate those levels of quality that are equal to average, or better than average, of those known to exist in the state of the art and should permit the adoption of higher levels of measurement consistent with normal progress and development.

A study of some of the quantitative limits permitted in several of the standards listed in the author's paper does not indicate that this premise has been observed. The establishment of mediocre levels of performance or construction is not conducive to the improvement of the art, nor does it represent real engineering progress.

It is to be hoped that in future efforts in the establishment of test codes and standards, we will not set our sights too low.

FERDINAND JEHLE, Indianapolis, Ind.: It seems to me that as far as these codes go, they look all right, but they are limited to only the firing devices of a heating plant. We can have good firing devices and still have a pretty bad plant. It seems to me that codes should be extended to some of the other things that are necessary in heating plants, such as controls, thermostats, air valves, traps, pumps, and the like. Unless all of these things meet some kind of a standard, I do not believe the plant as a whole will be satisfactory. I think maybe Mr. Thulman hinted at that. I do not know whether he has the same gadgets in mind as I do or not.

MR. THULMAN: I did not think I was hinting.

MR. JEHLE: Well, you did not name the gadgets.

L. E. SEELEY, New Haven, Conn.: I wonder if Mr. Dill could point out how many of these codes materialized within the last five or ten years? I suspect that this list of codes would reveal that many of them have materialized rather recently and those who have worked on codes 20 years ago realize how hard it was to even get one. I wonder if this development of codes is not moving along at an accelerated pace, and that the items people have pointed out as being desirable will be attained. We are just getting up speed, I think. If you could tell us that, I would like to hear it.

MR. DILL: I cannot answer Professor Seeley's question, especially about the codes covering the gas-burning devices. I do not know how long they have been in effect I think all of the commercial standards in the papers marked *T.S.* are certainly younger than five years old with the possible exception of that one on anthracite stokers. I am not sure about that. The I=B=R codes, I believe, are not five years old as yet.

1273

No. 1273

THE INFLUENCE OF HEAT CAPACITY OF WALLS ON INTERIOR THERMAL CONDITIONS AND HEAT ECONOMY

C.-E. A. Winslow,* L. P. Herrington** and R. J. Lorenzi,† New Haven, Conn.

OBJECT *And Nature of Experiments.* The purpose of these studies, which have been conducted for the past two years at the John B. Pierce Laboratory of Hygiene, was to measure the influence of walls of moderate and of high heat capacity upon interior thermal conditions and fuel consumption.

The original experimental plant consisted of a two-room house of ordinary frame construction, surrounded by shell spaces, on all four sides and above the roof and below the floor, in which spaces temperatures could be maintained at any desired point, to simulate external weather conditions. The structure was one story, the two rooms being separated by an interior corridor four feet wide.

After completion of studies in the original frame rooms, the experimental house was reconstructed—one room with a high-capacity wall of 16 in. of brick, the other with a wall of very low heat capacity. The series of studies made on the frame structure have now been repeated with the brick room, and it is the comparison of results with these two wall materials which is presented at this time.

The general layout of the original experimental installation may be found in a paper by Winslow, Greenburg, Herrington and Ullman.[1]

The frame room had an interior length of 15 ft (east and west walls); a width of 12 ft (north and south walls); and a height of 8 ft. The walls on their inside surface were of ¾-in. painted plaster on wood lath. The laths were nailed to 2 in. x 4 in. stud framing, the studs being placed 16 in. on centers. The exterior surface was 1 in. of wood sheathing covered by 1 in. of painted cedar clapboards. In the south wall there was one window and in the east wall there were two windows. The windows were double-hung wood, with metal weather-stripping, 2 ft 6 in. x 4 ft 8 in., and had storm sash weather-stripped with ⅛-in. felt. The west wall (abutting on the interior corridor) was a standard inside partition of wood lath with ¾ in. of plaster on each side of 2 in. x 4 in. studding, with 1 in. of insulating board added to the interior surface. The room had a double floor, an upper surface of 1 in. hardwood and a lower 1 in. of sub-flooring with a layer of heavy felt

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[1] Design and Equipment of the Pierce Laboratory, by C.-E. A. Winslow, Leonard Greenburg, L. P. Herrington and H. G. Ullman. (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 67.)

Presented at the 51st Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, Mass., January, 1945.

between. This double floor was nailed to 2 in. x 12 in. joists, 16 in. on center, providing a space of 12 in. between the bottom of the floor and a layer of metal lath and plaster separating this under-floor space from the controlled shell space beneath. During our experiments the floor was covered with a 12 ft x 15 ft broadloom rug with a pad underneath, each about ⅜ in. thick. The ceiling of the room was of wood lath and ¾ in. plaster on 2 in. x 4 in. ceiling joists set on 21-in. centers. Above the lath and plaster were 4-in. batts of rock wool, and above the rock wool was a 34-in. attic space, separated from the controlled shell space above by a roof of 1-in. sheathing on 2 in. x 8 in. rafters, 21 in. on center.

The brick room was exactly the same except for the reconstruction of the three exterior walls and except for the fact that the western instead of the eastern room of the house was used for the study of brick reconstruction. Therefore, the wall with two windows was, in this case, the west wall and the wall of the interior corridor was the east wall. The brick room was designed so that its interior dimensions were almost exactly the same as those of the frame room (actually 15 ft 1 in. x 11 ft 10 in. x 8 ft). On the three exterior walls, ¾-in. painted plaster was applied directly to a 16-in. wall of brick. The inner 12 in. were of medium density brick, the outer 4 in. of medium high density brick. Since the interior dimensions were the same as those of the frame room, the brick structure projected out for a considerable distance into the north, south and west shell spaces, and the windows were set in alcoves when viewed from the shell space side. The exterior surface exposed to the controlled shell spaces was thus notably increased.

Since exact knowledge of heat input was essential for our purposes, electrical heating was used. The experimental rooms were heated by radiation from a series of banks of 250-w infra-red lamps with silver-necked reflectors, having filament temperatures at full wattage of 4000 deg. They were mounted in sheet metal reflectors, suspended from the upper part of the walls with the bases of the heater bulbs 7 ft above the floor. At the north and south ends of the room was one bank of 6 bulbs; on the east and west sides, three banks with a total of 14 bulbs for each side. The lamps on the north and south walls, and the center banks on the east and west walls were directed toward the floor, while the four end banks on the east and west walls were directed toward the ceiling, so that heat actually reached the rest of the room by radiation from the floor and ceiling.

Heat input was regulated by automatic control of voltage actuated by a thermostat whose element was installed in a black globe.

During the course of an experiment, continuous temperature records were made on Leeds and Northrup multiple point potentiometer recorders (micromax type) from 32 recording points, which represented 130 couples of iron-constantan placed in the various shell spaces and in the room. They included, in most of the experiments here reported, 7 points in the air of the various shell spaces, 12 on the outside and inside wall and floor and ceiling surfaces, 2 on inside and outside window surfaces, 3 air and globe temperatures within the room, 6 points on a gradient through the north (windowless) wall (from inside air to outside air), and 2 ice bottle controls. In many instances, a single recording point averaged the temperatures registered by 4-8 thermocouples distributed over a given surface. On the interior surfaces of both rooms, the couples were imbedded in the surface of the plaster, and on the

outside of the brick structure the junctions were placed in grooves 1/16 in. to ⅛ in. deep and 6 in. long. On the outside clapboards of the frame structure, however, the thermocouples were placed on the surface of the clapboards.

In general, experiments were made with a constant temperature (of 20 F and 40 F) in the three exterior shell spaces or else with a varying temperature in the shell spaces (10 F-30 F or 30 F-50 F), duplicating a diurnal cycle. In most instances the interior room temperature was stabilized at 70 F and, after this temperature was maintained at this leved for 8 hours, the heating units were turned off and kept off for 8 hours to simulate a period of night-cooling. Where the outside temperature was cycling, this period of night-cooling was synchronized with the coolest portion of the exterior cycle. In one experiment with the brick room, the windows were opened during the period of night-cooling. At the close of the cooling phase the heating units were turned on again and comfort conditions restored as promptly as possible.

Experiments were also made to reproduce summer conditions by varying the temperature of the three exterior shell spaces between 65 F and 95 F and back again in a regular diurnal cycle. In this case, of course, no heat was supplied to the room.

Finally, for some special studies on heat conductance and heat capacity, the interior of the room and all shell spaces except that outside the north wall were maintained at 90 F while the north shell space itself was kept at 10 F or 30 F. In this way we obtained a major temperature differential, with heat-flow only through the windowless north wall.

Heat Conductance and Heat Capacity of Wall Materials. With stable conditions inside and outside the experimental room, and with exact knowledge of the wattage input necessary to maintain such a stable relationship, it was possible to compute conductance values for the types of building materials involved on a larger and more practical scale than that provided by ordinary panel tests. Five satisfactory experiments were completed with this end in view for the frame and six for the brick structure. The interior temperature was 70 F and the outside temperature 40 F in Experiments Frame V and Brick I-A. The interior temperature was 70 F and the outside temperature 20 F in Experiments Frame VI and VII and Brick II. In these instances all three exterior shell spaces were cold. In Experiments Frame XIII and XIV and Brick V all temperatures except that of the north shell space were at 90 F, while the north shell space was at 10 F, 15 F, and 30 F, varying in the different experiments. This setup should, of course, be the most accurate since the north wall has no window and since heat interchanges through other surfaces were relatively slight.

For a given period of five experiments with the frame structure and for six different periods in three experiments with the brick structure, the exact wattage required to maintain a differential of 1 deg between inside and outside air and between inside surface and outside surface was known. From the total amount of heat supplied to the room were subtracted the computed losses by leakage and computed losses by conductance through windows, floors, ceiling and interior wall (and in the case of Experiments Frame XIII and XIV and Brick V, also through the two exterior walls not subjected to cooling). Appropriate conductance values given in the HEATING, VENTILATING, AIR CONDITIONING GUIDE, corrected for actual shell space air velocities, were used in these computations; from the nature of the experiments, these losses were

relatively small. The total wattage input less the losses by windows, floor and ceiling conductance gave the flow of heat through the wall structure under study. The results of these eleven experiments are summarized in Table 1.

The first and fourth columns of figures in this table show the theoretical conductances of the frame and brick structures as computed from the tables in THE GUIDE. The third and sixth columns show the conductance values actually observed when computed on the internal surface area of wall. It will be noted that, in the case of the frame structure, these values are 23 per cent in excess, and in the case of the brick structure, 34-37 per cent in excess of the theoretical value. It is, however, from its outer surface that a building loses heat, and while the difference between inside and outside surface is relatively slight in a large structure, it is considerable in a two-room house. In the second and fourth columns, computed conductance figures were based on the mean between the inside surface area and the corresponding total outside surface area exposed to cooling (including, of course, the overlap at the top of the sides and around the windows in the case of the brick room). These figures check closely with theoretical values, being very slightly higher in the case of the frame and practically identical in the case of the brick structure. The general mean figures for all experiments, when computed on mean wall area, are as follows:

	Coefficients Btu/(hr) (sq ft) (° F)	
	Air-to-Air	Surface-to-Surface
Frame: Theoretical	0.22	0.30
Observed	0.24	0.32
Brick: Theoretical	0.29	0.43
Observed	0.29	0.44

Thus, the brick wall has a surface-to-surface coefficient about 37 per cent higher than that of the frame wall.

Heat capacity was determined for the frame structure from the data of Experiment Frame XII, in which the outside shell spaces were cycled between 65 F and 95 F for two complete cycles. Four periods were selected, during each of which heat was flowing in the same direction through both interior and exterior surfaces. For each of these periods the amount of heat transferred between shell space and outside wall surface (per square foot) and the amount of heat transferred between room air and inside wall surface was estimated from the film coefficients given in THE GUIDE (1.75 for outside frame surface and 1.52 for inside plaster wall). The net excess heat flow into or out of the wall for the period was then divided by the net change in wall temperature in the same period (determined by the mean of the change in inside wall temperature and outside wall temperature during the period). This procedure gave values ranging from 2.0 to 3.0 with a mean of 2.6 Btu per (sq ft) (deg F change in temperature). This compared with a theoretical figure of 2.56 for a wall of generally corresponding structure.

A similar analysis was made of the results in Experiment Brick VI with the brick structure which yielded a figure of 37.6. The soundness of this pro-

cedure was considered doubtful because of the fact that THE GUIDE coefficients are based on surface couples and those used by the authors, in the case of the brick structure, were both imbedded. In the case of the frame room, the error due to the interior imbedded couple could not be quantitatively great. For the brick room a more accurate approach was obtained from the data of Experiment Brick V. Here there were two conditions of stability; first, with an exterior temperature of 14 F, and second, with an exterior temperature of 40 F—the interior of the room being kept at 90 F throughout. Between these two stable phases there was a heating-up period of 67 hours, which gave an admirable measure of heat capacity. In this case, however, it was first nec-

TABLE 1—CONDUCTANCE VALUES

Btu/(hr)(sq ft)(° F)

EXPERIMENT	AIR-TO-AIR COEFFICIENTS[a]			SURFACE-TO-SURFACE COEFFICIENTS		
	Theoretical	Observed		Theoretical	Observed	
		Based on Mean Surf.	Based on Int. Surf.		Based on Mean Surf.	Based on Int. Surf.
Frame V	0.22	0.25	0.29	0.30	0.33	0.38
VI	0.22	0.25	0.28	0.30	0.33	0.38
VII	0.22	0.24	0.28	0.30	0.32	0.37
XIII	0.22	0.24	0.27	0.30	0.32	0.37
XIV	0.22	0.22	0.25	0.30	0.29	0.33
Brick I-A						
Phase 1	0.29	0.29	0.39	0.43	0.44	0.60
Phase 2	0.29	0.30	0.41	0.43	0.46	0.62
Brick II						
Phase 1	0.29	0.29	0.40	0.43	0.46	0.62
Phase 2	0.29	0.30	0.41	0.43	0.46	0.63
Brick V						
Phase I	0.29	0.29	0.37	0.43	0.44	0.55
Phase 3	0.29	0.28	0.35	0.43	0.42	0.53

[a] Based on shell-space air velocity of 100 fpm and still air inside.

essary to determine actual film coefficients for the inside and outside walls under the conditions of the experiment. It was found that the outside coefficients in THE GUIDE, in this case, yielded quite unreasonable results. This is to be expected since the film coefficients are not only based on surface thermocouples, but also on a wind velocity of 15 mph. In the case under consideration, the fact that the couples were imbedded in the walls would tend to give lower coefficients. In Experiment Brick V, too, the inside wall was warmed by radiation from the warm ceiling and walls, and this would tend to raise the interior coefficient, while the outside surface was radiating to the cold outer wall of the shell space, which would tend to lower the exterior coefficient.

The actual film coefficients obtained in the experiments were computed for the two stable phases of Experiment Brick V from conductance values from surface to surface (0.43), and from air to air (0.29), as in Table 1, and from the actual temperature differential from outside surface to inside surface, and

from outside air to inside air. The mean values obtained were 1.49 for the outside film coefficient and 1.55 for the inside film coefficient. Using these new film coefficients, the heat transfer from the outside wall to the shell space for the 67 hours of heating-up was subtracted from the heat transfer from room air to inside surface and divided by the temperature rise of the wall (mean of inside and outside surface) to give the heat capacity per square foot of wall. The resulting value was 33.4. The value of 37.6 obtained by a cruder

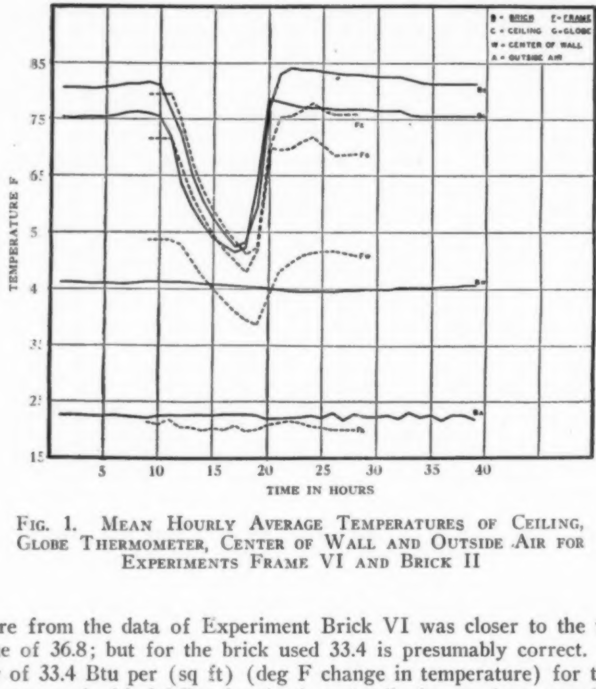

FIG. 1. MEAN HOURLY AVERAGE TEMPERATURES OF CEILING, GLOBE THERMOMETER, CENTER OF WALL AND OUTSIDE AIR FOR EXPERIMENTS FRAME VI AND BRICK II

procedure from the data of Experiment Brick VI was closer to the theoretical value of 36.8; but for the brick used 33.4 is presumably correct. A heat capacity of 33.4 Btu per (sq ft) (deg F change in temperature) for the brick wall as compared with 2.6 Btu for the frame wall, shows a heat capacity about thirteen times as great with only a 37 per cent greater conductance value.

Comparative Performance of Frame and Brick Walls with a Constant External Temperature of 20 F: A few typical experiments may be cited to illustrate the comparative performance of the two structures studied. Fig. 1 presents mean hourly average figures for Experiment Frame VI and Experiment Brick II, obtained at four of the 32 points at which temperatures were recorded. The two upper curves represent the temperatures of the ceiling surface from which most of the heat was radiated to the room. The second pair of curves represent the mean globe temperatures in the center of the room. The third pair represent the temperatures in the center of the actual wall structure itself (windowless north wall). The lowest curves represent the temperatures of the air in the shell space outside this north wall.

It will be noted that the shell space temperature varied between 22 and 23 F in the case of the brick structure, and between 20 and 22 F in the case of the frame structure.

During the first stable phase, the ceiling was between 79 and 81 F in both instances, while the mean globe temperature was 75.5 F for the brick and 71.5 F for the frame room. This difference in globe temperature was due to the fact that in the frame experiment the shell space was a little cooler and the ceiling temperature a little lower.

Globe temperatures in the frame room represent the mean of two globe thermometers at 30-in. and 60-in. levels; in the case of the brick room, a single reading was obtained by averaging the temperatures recorded by four globe thermometers at levels of 6 in., 30 in., 60 in., and 90 in. above the floor. Special check readings showed that when the heating lights were on, the high temperatures of the 90-in. globe raised the general mean about 1 deg above the mean of the 30-in. and 60-in. levels. Therefore, in the diagrams here presented, the globe values for the brick room, when the heat is on, are about 1 deg too high for accurate comparison with the globe values for the frame room.

Between the 10th and the 11th hours for the brick room (the frame experiments began only on the 9th hour of the schedule on which Fig. 1 is plotted), the lights were turned off for the period of night-cooling. In the next 7 hours when the lights were off, the globe temperature dropped 23-24 deg in both experiments.

When the heating units were turned on again (between the 17th and 18th hours for the brick room and one hour later in the corresponding cyclical phase for the frame room), the temperatures rose rapidly. By the 20th hour the globe temperature of the brick room had reached a maximum of over 78 F, falling later to a level of 75.5 F, the same as its initial value. The globe in the frame room rose more slowly (less heat being applied) to 72 F at the 24th hour, falling later to a level of 68.5 F, 3.5 deg lower than its initial value. The heat input to the frame room at the close of this experiment was not quite enough to cause it to reach the initial temperature.

The curves for the center of the wall structure are of special interest. It will be noted that in the frame structure the center of the wall (mean of those surfaces of the interior lath and of the exterior sheathing facing each other across the stud space) fell from 53.5 F to 38.5 F and rose again to 52.0 F (the heat capacity not being fully satisfied at the end of this experiment, as pointed out). In the case of the brick room, on the other hand, the temperature at the center of the brick wall fell only from 46 to 44.5 F, rising again to 46 F at the close. Practically the entire change in room temperature during the cooling period was absorbed by the interior half of the brick structure.

Experiment Brick II, as discussed previously, was exactly duplicated in Experiment Brick VII, except that during the night-cooling phase the top sash of one of the windows on the west wall was lowered 8 in. and the bottom of the south window was raised 8 in. Both experiments were very successfully controlled, the globe temperatures being 75.5 F for the closed-window room, both in the initial and final phases, and for the open-window room, 72.5 F during the initial and 72.0 F during the final phase. Both experiments were sufficiently prolonged to insure final stability of room conditions, including an

8-hour preliminary stable phase, 8 hours of cooling, 5 hours of heating-up, and 12 hours for reaching stability.

The effect of the more rapid cooling due to the open windows on globe temperature is indicated in the tabulation given below:

GLOBE TEMPERATURE (DEG F) IN BRICK ROOM WITH WINDOWS CLOSED AND WINDOW OPEN DURING NIGHT-COOLING PHASE

	Initial Stable Value	After First 3 Hours of Cooling	Minimum	After First 3 Hours of Heating	Maximum	Final Stable Value
Windows Closed	75.5	59.5	51.5	78.5	78.5	75.5
Windows Open	72.5	41.5	37.0	73.5	73.5	72.0

TEMPERATURES (DEG F) AT VARIOUS POINTS ON COOLING CYCLE

Structure	Air Temperature External Shell Space	Mean Globe Temperatures, Center of Room		
		Initial Stable Phase	After 3 Hours of Cooling	Minimum
Frame	20	71.5	57.5	48.0
	12–31	72.5	60.5	47.5
Brick	20	75.5	59.5	51.5
	15–33	75.0	59.5	51.5

With the windows closed, the globe temperature fell 16.0 deg during the first three hours of cooling and finally reached a minimum of 51.5 F; with the windows open, the fall for the first 3 hours was 31.0 F, and the final minimum, 37.0 F. The greater heat supplied by the regulators during the first three hours of heating-up raised the globe temperature in the closed-window room 27.0 deg and in the open-window room 36.5 F. This is an interesting example of the flexibility of the system of heating employed. The minimum globe temperature was 37.0 F, as shown. The heating units were turned on at a little before 8 a.m. The globe temperatures (deg F) recorded for the subsequent 15-minute intervals were as follows:

8:00	39.5
8:15	51.0
8:30	65.5
8:45	71.5
9:00	74.5

Heating of this sort can be turned on and off like a light, and rooms allowed to cool at night can be made comfortable in half an hour.

There was, however, one other important result of the open-window experiment. With windows closed, the night-cooling phase reduced the temperature

at the center of the brick wall from 46.0 F to 44.5 F, and at the close of the experiment this temperature had risen again to within a fraction of a degree of its original value. With the windows open, on the other hand, the temperature of the center of the brick wall fell from 46.5 F to 42.5 F, and at the close of the experiment—after 22 hours of heating—this value was still below 44.0 F. The high heat capacity of the brick wall causes it, as it were, to store cold during the phase of night-cooling. The effect of this accumulated heat

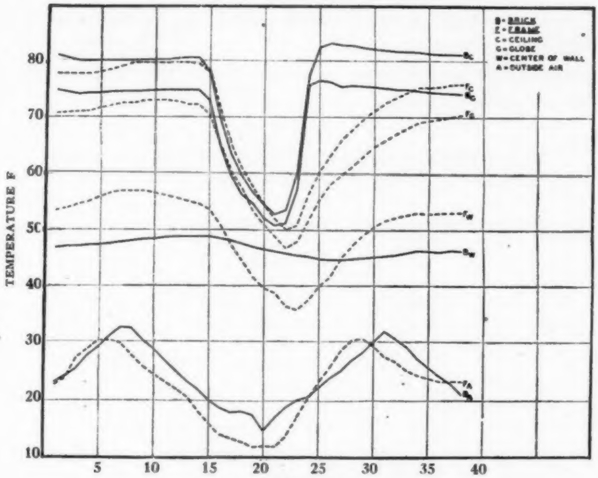

FIG. 2. MEAN HOURLY AVERAGE TEMPERATURES OF CEILING, GLOBE THERMOMETER, CENTER OF WALL AND OUTSIDE AIR WITH EXTERNAL TEMPERATURE PASSING THROUGH REGULAR DIURNAL CYCLE

deficit upon heat economies to be effected by night-cooling will be discussed in a later paragraph.

Comparative Performance of Frame and Brick Walls with Cycling External Temperature: Fig. 2 shows the data obtained with an external temperature passing through a regular diurnal cycle, the night-cooling phase being timed to come at the low point of the cycle. The chief purpose of this study was the determination of the economy effected by night-cooling under such conditions, but a few words may be said in general comments on the results obtained.

It will be noted that the external shell-space temperature started in both experiments close to 23 F. Over a period of 5-6 hours, this temperature was raised to 33.0 F for the brick room and 31.0 F for the frame room. Then, over a period of 13-14 hours, it was dropped to 15.0 F for the brick room and 12.0 F for the frame room. It was then raised, over a period of 8-11 hours,

to 32.0 F for the brick room and 31.0 F for the frame room, and finally lowered again to about 23.0 F for both rooms.

Fig. 2 shows how the interior cooling phase was superimposed on this exterior weather curve. It should be noted, however, that the study on the frame room was conducted in an early period of the studies, when the heating-banks and regulating system were designed to produce a slow and gradual heating-up after the cooling phase. The difference in the heating-up phase between the two structures was due to this difference in technique and not to any intrinsic difference between the brick and frame walls. The course of the cooling phase is almost identical in the constant and in the cycling experiment as in the tabulation at the left.

The change in the temperature of the center of the wall structure may be summarized as follows:

Temperature at Center of Walls at Various Points

Structure	External Shell Space	Interior of Room		
		First Maximum	Minimum	Second Maximum
Frame	20	53.5	38.5	52.0
	12–31	57.0	36.0	55.5
Brick	20	46.5	44.5	46.0
	15–33	49.0	45.0	46.5

The swing between maximum and minimum center-wall temperatures for the frame structure was 15 deg with constant outdoor temperature, and 21 deg when the interior cooling phase coincided with a low part of an exterior diurnal cycle. The corresponding swings for the center of the brick wall were 2.0 and 4.0 deg.

Heat Economies Accomplished by Night-Cooling: One primary objective in this study was to determine the economy of fuel which could be attained by turning off heat during a period of night-cooling, as related to the heat capacity of the wall structure. It was assumed that during this cooling period of eight hours the inside temperature could be allowed to drop to 50 deg without danger of local freezing of plumbing, and the control mechanism provided for turning on enough heat during this phase to avoid dropping materially below that figure.

During the actual period of cooling there should be a saving of one-third of the total daily heat input needed for a given outside temperature, but, obviously, additional heat must be supplied during the subsequent heating-up phase to satisfy the heat capacity of air, furniture, and structure. The balance between these two factors has been worked out in Tables 2 and 3.

Table 2 summarizes the computation of the excess heat needed to compensate for night-cooling with the frame and the brick structure under different atmospheric conditions (20 F, 40 F, cycles between 10 F and 30 F, and cycles between 30 F and 50 F). Lines 3-8 of Table 2 show the steps leading to

Table 2—Summary of Actual Power Requirements

	Frame Structure				Brick Structure			
1. General conditions	40F*	30–50F	20F	10–30F	40F	30–50F	20F	10–30F
2. Experiment number	I	XI	VI	IX	I–A	IV	II	III
3. Hours, stable phase	4–17	...	1–2	...	1–9	1–7	1–10	1–7
4. Mean watts, stable phase	1029	...	1580	...	1360	1330	2269	2256
5. Mean inside air, stable phase	69.3	...	69.7	...	71.3	70.6	74.1	73.8
6. Mean outside air, stable phase	37.9	...	21.2	...	40.3	40.4	21.9	20.7
7. Temp. differential	31.4	...	48.5	...	31.0	30.2	52.2	53.1
8. Watts/° F differential	32.8	32.7	32.6	32.7	43.9	44.0	43.4	42.5
9. Hours, heating & 2nd stable phase	...	24–41	10–19	22–38	18–49	29–44	19–44	30–45
10. Total watt-hours, heating & 2nd stable phase	...	17190	19806	25840	49650	27033	68867	43728
11. Mean inside air, heating & 2nd stable phase	...	64.8	65.1	62.3	71.0	70.9	74.4	72.4
12. Mean outside air, heating & 2nd stable phase	...	42.9	19.8	24.7	40.2	44.7	21.6	23.8
13. Temp. differential	...	21.9	45.3	37.5	30.8	26.2	52.8	48.6
14. Theoretical watt-hours heating & 2nd stable phase	...	12890	14768	20902	43268	18445	59580	33048
15. Excess watt-hours to satisfy heat capacity (10–14)	3443	4300	5038	4938	6382	8588	9287	10680

* The stable phase, in this case, was not at the beginning of the experiment which began with heating-up.

determination of the wattage necessary to maintain 1 deg differential between inside and outside air under stable conditions. In Experiments Frame IX and Frame XI no stable phase was maintained, and the figure in line 8 for these experiments represents the mean of the values obtained in Experiments Frame I and Frame VI.

Line 10 gives the total watt-hours actually supplied for the hours of the heating-up and second stable phase (line 9.) Lines 11-13 give the mean temperature differential existing between inside and outside air during this period, and line 14 the theoretical watt-hours which would have been required to maintain stability during this phase, as determined from lines 8, 9, and 13. By subtracting the theoretical watt-hours (line 14) from the actual watt-hours (line 10), the result is the excess watt-hours consumed in the satisfaction of heat capacity (line 15).

It is realized that in the case of the brick structure at least, the figure obtained in line 15 is somewhat less than the true value, since even after two cycles had been completed, the temperature of the wall structure was still falling slightly. The error due to this cause can hardly, however, be large. In

Table 3—Estimate of Saving by Night-Cooling

	Frame Structure				Brick Structure			
1. General conditions	40F	30–50F	20F	10–30F	40F	30–50F	20F	10–30F
2. 24-hour differential	30.0	30.0	50.0	50.0	30.0	30.0	50.0	50.0
3. Total watt-hours, without night-cooling	23616	23544	39120	39240	31608	31680	52080	51000
4. Temp. differential, stable phase	30.0	26.5	50.0	46.5	30.0	26.5	50.0	46.5
5. Total watt-hours, stable phase (16 hours)	15744	13865	26080	24328	21072	18656	34720	31620
6. Excess watt-hours, cooling phase	0	0	0	480	0	0	990	1040
7. Excess watt-hours, heating-up phase	3443	4300	5038	4938	6382	8588	9287	10680
8. Sum of (5), (6), and (7)	19187	18165	31118	29746	27454	27244	44997	43340
9. Per cent saving by night-cooling	18.8	22.8	20.5	24.2	13.1	14.0	13.6	15.0

Experiments Brick I-A and II the final stable values for brick-plaster contact and center-of-brick were less than 1 deg below the preliminary figure.

Table 2 represents the direct results of the experiments. In Table 3 an estimate was made of the watt-hours saving accomplished by a period of night-cooling under four specific external air conditions with each structure on the basis of the data in Table 2.

Line 3 of Table 3 shows the watt-hours required to maintain (under conditions of line 1) an interior temperature of 70 F for 24 hours (line 8 of Table 2 multiplied by line 2 of Table 3 and by 24).

In line 4 are indicated the mean temperature differentials maintained between inside and outside air during the 16 hours of the stable phase preceding the following night-cooling. For the experiments with constant external temperature, the figure is, of course, the same as that in line 2. With outside cycling, the figures in line 4 are computed as the difference between 70 F and the mean exterior temperature (in a cycle between 10 and 30 F or 40 and 50 F) in that part of a regular symmetrical cycle which begins four hours after the minimum and extends to four hours before the next minimum. The assumption is that the eight hours of night-cooling would cover the four hours on each side of the minimum point.

In line 5 the number of watt-hours needed during this interior stable phase is obtained by multiplying line 8 of Table 2 by line 4 of Table 3 and by 16.

In line 6 are given the additional watt-hours actually supplied during the cooling phase (to maintain an interior temperature not below 50 F). Line 7 shows the additional watt-hours required for heating-up (as derived in line 15 of Table 2). Line 8 is the sum of lines 5, 6, and 7. Line 9 gives the percentage saving accomplished by night-cooling, as computed from lines 3 and 8.

It will be noted that, contrary to views which are often expressed, the practice of permitting the structures studied to cool down to a minimum of 50 F at night effects substantial fuel economy. The economy increases with the coldness of the external environment (20 F as compared with 40 F), and it is greater when the cooling is permitted at the trough period of a cycling outdoor temperature than when it occurs during a period of constant outdoor temperature (10-30 F and 30-50 F, compared with 20 F and 40 F).

The saving ranges from 18 per cent to 24 per cent for the frame structure, and from 13 per cent to 15 per cent for the brick structure.

During the cooling phase the same direct saving of the heat supplied is effected with either wall, and both walls become chilled—the frame wall even more than the brick wall. In the case of the brick, however, there is a much greater heat capacity to be saturated when the heat is turned on again, and in this process more of the heat saved must be expended during the warming phase.

Two factors must be taken into account in considering possible practical applications of these findings. In a large structure the factor of heat capacity (other than in walls) might be great, and economy by night-cooling would be correspondingly reduced. Furthermore, the method of heating employed was such as to avoid the losses of efficiency which accompany the use of many heating systems on an on-and-off basis.

The Influence of Heat Capacity on Comfort Conditions in Hot Weather. It is obvious that the heat capacity of wall structures has a possible influence

on the degree of protection afforded against high daytime temperatures in summer. Brief reference may be made to this point.

This type of observation is illustrated in Fig. 3 for Experiments Frame XII and Brick VI. In each case, external temperatures in the three shell spaces were varied in a regular diurnal cycle between 65 and 95 F—no heat, of course, being supplied to the interior of the room. The shell space below the

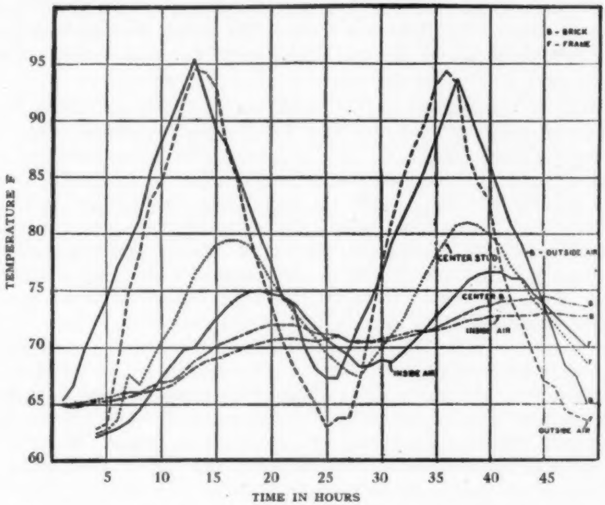

FIG. 3. INFLUENCE OF HEAT CAPACITY ON COMFORT CONDITIONS IN HOT WEATHER

floor was maintained at about 55 F to represent the cooler soil, and the temperature of the space above the ceiling was allowed to drift with the changes accompanying the diurnal cycle in the shell space.

Reference to Fig. 3 and to Table 4, which summarizes some of the salient points, will make the findings clear. It will be noted that tests were begun

TABLE 4—SUMMARY OF TEMPERATURES UNDER SUMMER CONDITIONS—DEGREES FAHRENHEIT

	Outside Air		Center of Wall		Inside Air	
	Frame	Brick	Frame	Brick	Frame	Brick
Start of experiment	63.0	65.5	62.5	65.0	62.0	65.0
Hour of 1st maximum	13	13	16	20	19	21
First maximum	94.5	95.5	79.5	72.0	75.0	71.0
Hour of 1st minimum	25	25	28	29	28	28
First minimum	63.0	67.5	67.5	70.5	68.5	70.5
Hour of 2nd maximum	36	37	38	45	41	45
Second maximum	94.5	93.5	81.0	74.5	76.5	73.0
Final figure	63.5	65.0	68.5	73.5	70.0	73.0

with all temperatures—inside and out—in the neighborhood of 65 F. The outside shell spaces were then cycled up to 95 F and down to about 65 F in two successive diurnal cycles. In the case of the brick room, the minimum attained at the close of the first cycle was 4.5 deg higher than that in the case of the frame house; otherwise the two experiments were closely comparable.

The temperature at the center of the wall structure followed that of the outside air with an interval of 2-3 hours in the case of the frame room, and 4-8 hours in the case of the brick room. The swing between maximum and minimum was about 30 deg for the outside air; 12.0 deg for the center of the frame wall, and 1.5 deg for the center of the brick wall.

The swing in temperature of the inside air (which followed by 1-3 hours the center-wall temperatures) was only 6.5 deg for the frame room and only 0.5 deg for the brick room. In the frame room the temperature fell at night to 68.5 F and rose to a maximum of 76.5; that of the brick room fell to 70.5 F and rose to 73.0 F. The second maximum was, in each instance, 1-2 deg higher than the first maximum, showing a slight heating-up of the total structure. Both walls give considerable protection against external daytime summer heat, but the brick room, with its maximum of 73.0 F, is, of course, most favorable. It is also important to note that the second maximum peak in the frame room came only five hours after the external peak period; while with the brick room the peak of inside air temperature came eight hours after the peak of external temperature—at a time when open windows could relieve the situation further. Obviously, these conclusions apply only to a climatic condition which involves a marked drop of temperature at night. Also, our results were obtained with a continuously cool under-floor space. It must also be pointed out that solar radiation did not enter into our experimental picture, and such radiation would, in itself, tend to create a greater inequality between day and night.

Clearly, the brick walls gave some added protection against external cycles of high temperature under the conditions of our experiment, but the surprising thing, perhaps, was the good performance of the frame structure. Even in the frame room an external diurnal maximum of 94.5 F was held down to 76.5 F.

SUMMARY OF CONCLUSIONS

1. Determinations of heat conductance and heat capacity values for frame and brick walls have been made in full-scale rooms under a variety of controlled experimental conditions. The conductance and capacity results for the brick wall check exactly with theoretical values, and for the frame wall deviate by less than 10 per cent from these results. In order to obtain significant conductance results with a two-room, one-story isolated structure, it is, however, necessary to use the mean areas of the exposed wall structures and not their interior areas.

2. The rooms were observed with continuous external temperatures of 20 F and with external temperatures cycling between 10 and 30 F. When interior heat sources were cut off for an 8-hour period of night-cooling, the inside

globe temperature dropped off at the end of 8 hours in both structures to a temperature of about 50 F. If two windows of the room were opened, 8 in. each, one at the top and one at the bottom, the minimum temperature reached was 37.0 F. The center of the frame wall fell from 53.5 to 38.5 F; the center of the brick wall from 46.0 to 44.5 F. Almost the entire change in room temperature was absorbed in the inner half of the brick wall. The electrical heating units (set up to provide reflective panel heating from ceiling and floor) were so efficient that at the close of a cooling phase in the open-window brick room, the globe temperature rose from 39.5 to 65.5 F in 30 min.

These studies were made with walls having the following characteristics:

	HEAT CONDUCTANCE, SURFACE-TO-SURFACE BTU/(SQ FT) (HR) (° F)	HEAT CAPACITY BTU/(SQ FT) (° F)
Frame Room	0.32	2.6
Brick Room	0.44	33.4

3. A series of eight comparable experiments indicated the saving of heat-input accomplished by an 8-hour period of night-cooling to 50 F after due allowance for the additional heat supplied to satisfy heat capacity demands. The saving was 13-14 per cent of the heating load needed to maintain 70 F for 24 hours in the case of the brick structure, and 19-24 per cent in the case of the frame structure. The saving becomes greater with lower external temperatures and is greater when heat is turned off in the cold phase of a diurnal cycle than in the case of constant outside temperature. The decrease of economy with a structure of high heat capacity is noteworthy.

4. Under summer conditions of high outside temperature, cycling between 65.0 F and 95.0 F, both wall structures gave a high degree of protection. Maximum outside temperatures of 94.0 F on such a cycle gave maximum globe temperatures of 76.5 F for the frame structure with a lag of 5 hours, and maximum globe temperatures of 73.0 F with a lag of 8 hours for the brick structure.

DISCUSSION

R. A. MILLER, Pittsburgh, Pa.: In the night eight-hour cooling cycle, since there was no exposure to the outside air, nor to the sky, nor to the sun during the daytime, how did you simulate the nighttime cooling loss?

DR. WINSLOW: The nighttime cooling loss was simply due to the loss to the shell space which was lowered at night to 65. Of course, as pointed out, we did not have the direct radiant heat of the sun in the daytime and we did not have the radiating loss to interstellar space at night. The test results show what would happen when changing outdoor air temperature from 65 to 95 and back again.

I think that those other two factors in practice would more or less operate in opposite directions, but obviously you can draw only limited conclusions from this experiment. It was, however, rather interesting to us to find how great was the protection afforded by the factor of high heat capacity, and I think that this has some bearing on the discussion of Professor Mackey's papers.[2]

[2] Summer Weather Data and Sol-Air Temperature—Study of Data for New York City, and Lincoln, Nebraska (see pp. 75, 93).

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No. 1274

RADIATION CORRECTIONS FOR BASIC CONSTANTS USED IN THE DESIGN OF ALL TYPES OF HEATING SYSTEMS

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Third progress report of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the University of California.

ALTHOUGH UNDERTAKEN primarily for the purpose of investigating radiant energy exchange as a factor in the design and analysis of panel heating and panel cooling systems, the cooperative panel heating research project has also provided an opportunity for systematizing and, in some cases, rationalizing the procedures used in heat load computations for all types of heating systems. Thus, while the more technical aspects of panel heating procedures, as developed in the various progress reports prepared on this cooperative project, are of limited interest to the designer who is not working with panel systems, there are, nonetheless, certain broad fundamentals of radiant exchange to which attention can be occasionally directed and which are of the utmost importance to every designer whether he be working with panel, radiator, convector, or warm air heating systems. While results of such general interest are necessarily of secondary importance to the main objective of the project, they are indicative of a frequently overlooked service which specialized research is sometimes able to afford to general advancement in a wide field. Thus, sponsorship of an investigation on a subject of restricted or of only potential interest may be justified, not merely through the probability of establishing new procedures which may be needed in the future, but also through the possibility of simplifying and of generalizing existing procedures in a way which will present greater effectiveness in their use *now*.

In all heating systems, of whatever type, three fundamental problems require consideration, as follows:

1. The energy exchange between occupant and his environment with emphasis on the convection-radiation relationship which is required to exist if comfort is to be realized.
2. The energy exchange inside the enclosure by convection between room air and solid surfaces and by radiation among the various types of surfaces which make up the enclosure.
3. The energy exchange between the exterior of the enclosure and its surroundings, the latter consisting of the ambient air, objects *seen* by the exterior surfaces (as the earth, other houses, the sun, etc.) and the sky.

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Each of the foregoing problems is given a highly conventional and, in some cases, inexact treatment in the standard load determination procedure which is used in designing usual types of heating systems. With panel heating, however, the added emphasis on radiant exchange requires that more careful consideration be given to each of these problems and that more precise methods be established for evaluating design coefficients. Once these new methods have been developed, they afford an obvious opportunity for investigating the accuracy and the adequacy of the conventional procedures and they also, in some cases, afford a means of revising or of limiting the accepted values of design coefficients. The intent of this report is to call attention to two specific instances in which consideration of radiant exchange will significantly alter accepted ideas of design coefficients and, further, to provide in convenient graphical form a simplified and generalized means of selecting a design value of the inside air temperature corrected for radiant interchange between the occupant and the enclosing surfaces. Each of the three items considered in this report appears under one or another of the three major problems which have been listed in the preceding paragraph. For convenience, therefore, and to identify each constant with the over-all design problem of which it is a part, the report has been prepared in three sections.

PART I. THE DESIGN INSIDE AIR TEMPERATURE

The development in this section is based on the assumption that the designer is desirous of establishing environmental conditions equivalent to those which would exist in a room with air at 70 F and all inside surfaces at the same temperature. If, for any reason, the system is designed to provide a basic temperature other than 70 F the results of this analysis can still be used, with accuracy adequate for all practical purposes, by altering the corrected design inside air temperature by an amount equal to the difference between the base design temperature and 70 F. Except in the case of rooms having no transmitting surfaces, the average surface temperature of the enclosure will differ from that of the room air. In a room heated by radiant means the air temperature will usually be less than the average surface temperature, while in rooms heated by cast-iron radiators, convectors, or warm air systems the air temperature will inevitably exceed the average temperature of the enclosing surface. When the outside temperature is low or when the enclosure has walls of high conductance, the inside surface temperature may be reduced to a point such that some radiant loss will occur from the occupant. Under such conditions an appreciable increase in the design inside air temperature will be necessary in order to reduce the convective heat loss of the occupant by an amount sufficient to provide compensation and thereby re-establish an equilibrium heat balance in the region of comfort.

The magnitude of the compensating inside air temperature increase has been investigated experimentally by Houghten and McDermott [1] and their results—based on 51 tests—are summarized in a graph.[2] From this figure the required

[1] A.S.H.V.E. Research Report, No. 946—Cold Walls and Their Relation to Feeling of Warmth, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 83).
[2] HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1944, p. 63.

design inside air temperature in a room having three exposed walls with 45 F inside surface temperature is 79 F, an increase of 9 F needed to compensate for the greater rate of radiant loss from occupant to cold walls. THE GUIDE data provide an excellent reference for roughly estimating the relationship between cold walls and the feeling of warmth, but they cannot effectively be extrapolated from the single experimental condition of three cold walls and do not provide a means of integrating the effect of various types of surface (with correspondingly different inside surface temperatures) which are likely to be found in an actual design problem. Further, THE GUIDE data are expressed in terms of inside surface temperature and hence, for use in practice, require an analysis by the designer involving computations based on the actual conductances and the design value of the outside air temperature.

The same problem can be solved in a general form and expressed in terms of variables more amenable to exact and rapid use by the designer if the comfort equation which is used in design of radiant heating systems is accepted as a criterion of effectiveness and used in establishing a rational basis for the solution of the cold wall problem in convective heating. Neglecting the differences which can be shown to exist between the mean radiant temperature and the weighted average inside temperature, t_s, an approximate comfort equation can be written in the form,

$$t_a = \phi\,(t_s) \qquad (1)$$

where $t_s = (\Sigma t_x A_x)/(\Sigma A_x)$; t_x is the temperature of any section of inside surface of area A_x over which the surface temperature does not vary appreciably; t_a is the design value of inside air temperature corresponding to which a heat balance on the occupant in the room will be the same as though air and walls were uniformly at 70 F. In order to attain a solution having the desired simplicity, it is necessary first to evaluate t_s in terms of t_a and in terms of the physical constants of the room, then to fix explicitly the functional relationship indicated by the symbol ϕ, and finally to prepare a graphical solution of Equation 1 from which t_a can be readily determined from known or readily calculable constants of the system.

The problem of expressing t_s in terms of t_a and of physical constants is a simple one and can be solved from the fundamental heat transfer equations. For steady state conditions the rate of heat flow to an inside surface must equal the rate of heat transfer through that surface so the equation can be written,

$$U_x A_x\,(t_a - t_o) = h_x A_x\,(t_a - t_x) \qquad (2)$$

where t_a and t_o are inside and outside design air temperatures, U_x is the overall coefficient of heat transfer for the section of area A_x, and h_x is the combined inside film coefficient for transfer by convection and radiation. Strictly speaking, h_x would not have the same value for all points inside the enclosure, but accepted practice [8] is to consider this coefficient as having the constant value of 1.65 Btu/(hr) (sq ft) (deg F). When this assumption is permissible, Equation 2 reduces to the form,

$$t_x = t_a - 0.606\,U_x\,(t_a - t_o) \qquad (3)$$

[8] HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1944, p. 91.

Then substituting from Equation 3 for t_x in the definition of t_s,

$$t_s = \frac{\Sigma\,[t_a - 0.606\ U_x\ (t_a - t_o)]\ A_x}{\Sigma\ A_x}$$

$$= t_a - 0.606\ (t_a - t_o)\left(\frac{\Sigma\ U_xA_x}{\Sigma\ A_x}\right) \qquad (4)$$

The last parenthesis of Equation 4 is an integrating term leading to a weighted average value of the over-all coefficient of heat transfer and can be used to define an equivalent over-all coefficient,

$$U_s = \frac{\Sigma\ U_xA_x}{\Sigma\ A_x} \qquad (5)$$

where U_s is the equivalent uniform over-all coefficient of heat transfer of a room which would have exactly the same heat load (and the same t_s) as the actual room. This equivalent-room concept is a valuable tool for exploiting in full the possibilities of simplifying existing procedures for heat load determination and for performance analysis of either convection or radiant heating systems. The most general expanded form of Equation 5 which is likely to be needed for any average room is,

$$U_s = \frac{A_g}{A_s}U_g + \frac{A_w}{A_s}U_w + \frac{A_f}{A_s}U_f + \frac{A_c}{A_s}U_c + \frac{A_i}{A_s}U_i \qquad (6)$$

where the subscripts g, w, f, c, and i refer to glass, exterior wall, floor, ceiling, and interior wall, respectively. The denominator, A_s, is the total inside surface area of the enclosure. In most cases the interior partitions are non-transmitting, so U_i becomes zero; where this occurs it is important to note that the A_s term must continue to include the partition area even though there is no longer an explicit ΣA_iU_i term in Equation 6.

Noting that the transmission heat load from the actual room is customarily calculated from the equation,

$$Q_t = \Sigma\ U_xA_x\ (t_a - t_o) = (t_a - t_o)\ \Sigma\ U_xA_x = (t_a - t_o)\ U_sA_s \qquad (7)$$

it is apparent that the designer can readily determine U_s from computations which must, in any event, be carried out in determining the heat load; thus, while U_s is a new term, it is not one which requires calculation in excess of that otherwise necessary. Then, substitution from Equations 4 and 5 back into the comfort Equation 1 gives,

$$t_a = \phi\ [t_a - 0.606\ (t_a - t_o)\ U_s] \qquad (8)$$

The next requirement is to fix explicitly the functional relationship between the air temperature and the average surface temperature. If, as noted before, the differences between mean radiant temperature and average surface temperature can be neglected for practical convective heating purposes (consideration of such differences is hardly justified in view of the inaccuracy inherent in use of a fixed value of h_x) then the problem reduces to one of evaluating the increase in air temperature needed to compensate for a 1 F change in the mean radiant temperature. Experimental evidence on this subject is not yet conclusive, nor is opinion undivided. Dr. C.-E. A. Winslow's work at the Pierce Laboratories and Dr. Thomas Bedford's experimental work in England both indicate that, in the comfort region, a 1 F change in mean radiant tem-

perature can be compensated by an opposite and equal change in air temperature. This relationship was likewise observed by the authors in the reflection room tests[4] which have been previously reported. From theoretical considerations it can be shown also[5] that the influence of mean radiant temperature would be expected to be not less than equal to that of air temperature.

The only experimental work done explicitly to evaluate cold wall effects was that of Houghten and McDermott,[6] which has been previously referred to. Their results have not been expressed in terms of a t_a *vs.* t_s relationship, but from the data and from the published description of conditions under which the tests were made it is possible to calculate the ratio of air temperature increase to average surface temperature decrease. Examples of such calculations are given.

Example: The 5 ft x 6 ft x 6 ft *cold wall* room used in the A.S.H.V.E. tests[7] had two 6 ft x 6 ft cold walls and one 5 ft x 6 ft. The total cold wall area was therefore 102 sq ft. Floor and ceiling were of lattice work and ventilation air passed in through the ceiling lattice and out through the floor lattice; the ventilation rate was 31,800 cfh so the surface temperatures of the floor and ceiling (though not given in the original report) can probably be taken as close to air temperature. Likewise, the temperature of the one 30 sq ft uncooled wall is not specified so will be taken as approximately room air temperature. Then for conditions in the test room,

$$t_s = \frac{90}{192}\,t_a + \frac{102}{192}\,t_c$$
$$= 0.469\,t_a + 0.531\,t_c$$

where t_a and t_c are the room air and cold wall temperatures.

Referring to the test data (A.S.H.V.E. GUIDE, 1944, Fig. 10, p. 63) for conditions equivalent to a room with air and walls at 70 F, the following values may be tabulated:

t_c	t_a	$t_s = 0.469\,t_a + 0.531\,t_c$	$(t_a - 70)/(70 - t_s) = \dfrac{\Delta t_a}{\Delta t_s}$
45	79	60.95	0.994
50	75.8	62.1	0.734
55	73.9	63.85	0.634
60	72.4	65.80	0.561
65	71.2	67.90	0.571
70.0	70.0	70.0	....

The gradual increase in the $\Delta t_a/\Delta t_s$ ratio as t_s decreases is interesting. From the standpoint of accuracy in physical measurement the data for low wall temperatures would be expected to be more accurate than for higher wall temperatures since the magnitude of the two differences which are used in calculating the ratio are much

[4] A.S.H.V.E. RESEARCH REPORT No. 1192—Panel Heating and Cooling Performance Studies, by B. F. Raber and F. W. Hutchinson (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 35).
[5] Radiant Energy Exchange as a Factor in Airplane Cabin Heating, by B. F. Raber and F. W. Hutchinson (*Journal of the Aeronautical Sciences*, Vol. 11, No. 3, p. 239, July, 1944).
[6] Loc. Cit. See Note 1.
[7] Loc. Cit. See Note 1.

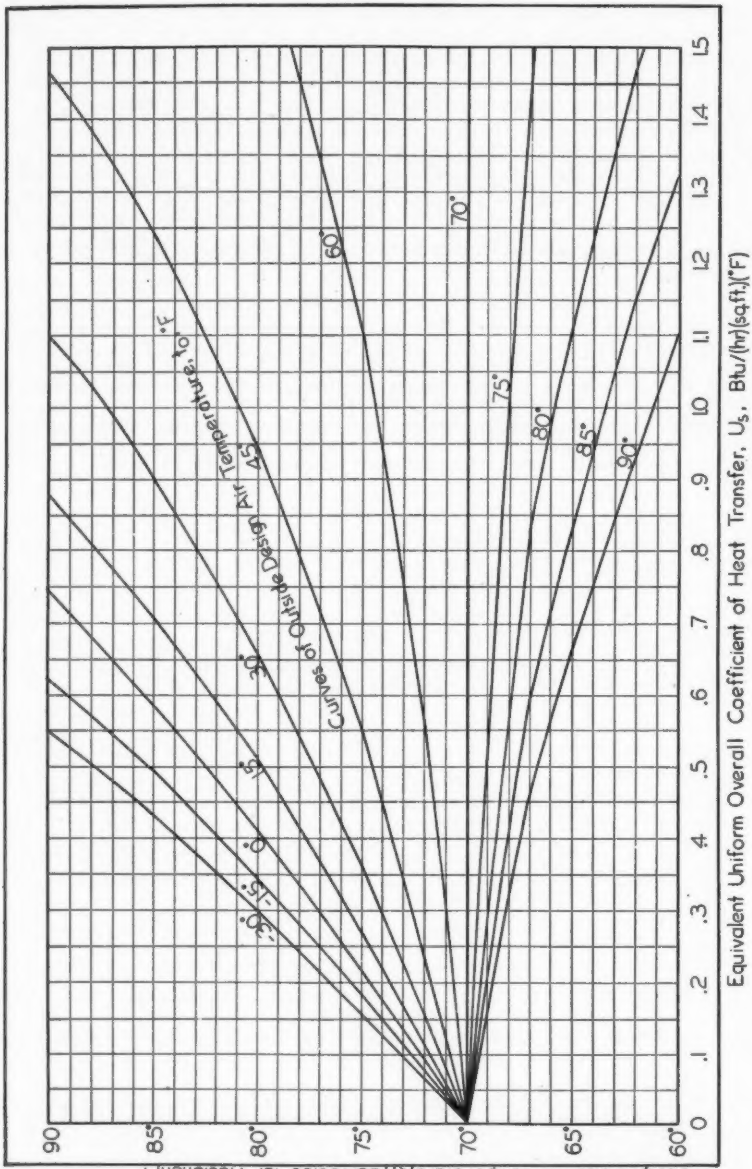

FIG. 1. GRAPHICAL EVALUATION OF COLD WALL EFFECT

greater for these conditions. On the other hand, the increased magnitude of the ratio may be due to changing influences of the assumption that the surfaces of the floor, ceiling, and uncooled wall were at room temperature.

In a more recent paper, Houghten[8] and coworkers find an even lower ratio, 0.3 deg ET (approximately 0.4 deg dry-bulb temperature) as a compensating air change for a 1 F change in MRT. Based largely on this latter report THE GUIDE (1944, p. 64) gives 0.5 deg ET (approximately 0.62 deg dry-bulb temperature) as the air temperature alteration needed for each 1 F change in MRT (mean radiant temperature).

Faced with the differences of data and of opinion as evidenced, an arbitrarily selected value of the $\Delta t_a/\Delta t_s$ ratio will be selected for the present purpose. Based on a study of all available data and with due allowance for the need of selecting a ratio which, if in error, will give a margin of safety, the authors suggest that the 1 to 1 ratio, in agreement with the findings of Dr. Winslow and coworkers and of Dr. Bedford and coworkers, be used. Such a value seems a feasible one, not only because there is a substantial body of data in support of it, but also because its use will give over-correction rather than under-correction of the air temperature. Using the 1 to 1 ratio and taking 70 F air with 70 F walls as a base condition, the comfort Equation 1 takes the very simple form,

$$t_a = 140 - t_s \qquad (9)$$

and Equation 8 becomes,

$$t_a = 140 - t_a + 0.606\,(t_a - t_o)\,U_s \qquad (10)$$

or,

$$t_a = \frac{140 - 0.606\,U_s t_o}{2 - 0.606\,U_s} \qquad (11)$$

Equation 11 gives the corrected design value of the inside air temperature in terms of the two design variables t_o and U_s. This equation provides a general solution of the cold wall (or, for summer conditions, the hot wall) problem and permits graphical interpretation in a form which can be used rapidly and simply by the designer. As an aid in establishing points for the graphical solution, Equation 11 can be written more conveniently in terms of U_s rather than t_a as the unknown,

$$U_s = \frac{3.3\,t_a - 231}{t_a - t_o} \qquad (12)$$

This equation is solved graphically in Fig. 1.

Example 1: A room 20 ft x 20 ft x 10 ft has two non-transmitting inside walls and two exterior walls ($U = 0.25$) each having a window ($U = 1.13$), the total room glass area being 113.6 sq ft. The floor has a U value of 0.25 (exposed to outside air) and the ceiling is non-transmitting. For an outside temperature of 0 F determine the design inside air temperature and calculate the increased load needed because of the excess radiant loss. By Equation 7,

$$\begin{aligned} Q_t &= (t_a - t_o)(113.6 \times 1.13 + 286.4 \times 0.25 + 400 \times 0.25 \\ &\quad + 400 \times 0 + 400 \times 0) \\ &= (t_a - t_o)\left[\frac{300}{1600}\right] 1600 \\ &= (t_a - t_o)\,0.1875 \times 1600 \\ &= (t_a - t_o)\,U_s A_s \end{aligned}$$

Then enter Fig. 1 at $U_s = 0.1875$, rise to intersect the curve for $t_o = 0$, move left to read $t_a = 74.2$ F.

[8] A.S.H.V.E. RESEARCH REPORT No. 1193—Radiation as a Factor in the Feeling of Warmth in Convection, Radiator, and Panel Heated Rooms, by F. C. Houghten, Carl Gutberlet, E. C. Hach (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 55).

The total transmission loss for comfort at $t_o = 0$ is, $Q_t = 74.2 \times 0.1875 \times 1600 =$ 22,260 Btu/hr, of which $4.2 \times 0.1875 \times 1600 = 1260$ Btu/hr is increased transmission loss needed to compensate for the cold wall effect; this amounts to a 6 per cent increase over the load needed to maintain a 70 deg air temperature.

Example 2: Consider that the room of the previous example is insulated sufficiently to reduce U_e to 0.10. In this event t_a, from Fig. 1, is found to be 72.2 F. The total heat loss from the room is now $72.2 \times 0.1 \times 1600 =$

$$11{,}550, \text{ a reduction of } \frac{22{,}260 - 11{,}550}{22{,}500} = 47.5 \text{ per cent.}$$

The result of *Example 2* is of particular interest because it brings out very clearly the saving in heat load (for equivalent comfort) which can be accomplished by insulation and the rather astonishing fact that the actual saving is greater than the value indicated from a comparison of over-all coefficients alone. Thus, for the case of these two examples, the savings due to reduced U-value is $\frac{0.1875 - 0.1000}{0.1875} = 46.7$ per cent against the actual heat load reduction of 47.5 per cent. The magnitude of the *dividend* due to insulation increases as the difference between the initial and final values of the equivalent over-all coefficient. In a room where the cold wall effect is not severe, insulation will reduce the heat load in the ratio of over-all coefficients, but in a room having marked cold wall characteristics the reduction in load will increase more rapidly than the reduction in equivalent over-all coefficients.

Consideration of the foregoing examples will bring out one way in which heating panels, or cast-iron radiators, or any other exposed heated surface can serve to reduce heat load on the system. Aside from the general problem of panel heating, suppose that the room of *Example 1* is to be heated basically by convection, but that a heating element, of whatever form, can be exposed in the room. Assume, further, that this heating unit will be placed near, but not against, an exterior wall so that it will blank out part of the wall from the occupant's view, but will not alter the inside wall surface temperature. Neglecting radiation, except insofar as it influences the design inside air temperature, and assuming that the exposed heating surface at 185 F consists of an outset base plate 1 ft in height and running along both exterior walls (area 40 sq ft); calculate the effect of this exposed plate on the design room air temperature and on the room heat loss.

The equivalent surface temperature (aside from the base plate) for the condition of *Example 1* is,

$$t_s = t_a - 0.606 \times 0.1875\, t_a = 0.8864\, t_a$$

With the base plate included, a new t'_s is established.

$$t'_s = t_s A_s + t_p A_p = \frac{0.8864\, t_a\, 1600 + 185 \times 40}{1600 + 40} = 0.865\, t_a + 4.5$$

Then, substituting this value in the comfort equation,

$$t_a = 140 - 0.865\, t_a - 4.5 = 135.5 - 0.865\, t_a$$

giving,

$$1.865\, t_a = 135.5, \text{or } t_a = 72.6 \text{ F.}$$

Thus, use of the base plate has reduced the heating load by

$$\left(\frac{74.2 - 72.6}{74.2}\right) 100 = 2.2 \text{ per cent.}$$

Whatever the output of the base plate may be, it partially meets the total convective load and hence does not require consideration in any particular way. By analogous reasoning it can be readily established that the installation of an inside partition in an underheated room with four exposed walls would increase the feeling of comfort for the same total energy input. Conversely, if a heating system were operating at the rate needed to maintain comfort in a room with four exposed walls, installation of an interior partition would permit realization of equal comfort with a reduced energy input.

PART II. THE INSIDE COMBINED FILM COEFFICIENT

The second of the three fundamental problems listed in the introduction is the one involving convective and radiant energy transfer within an enclosure. As previously mentioned, usual procedure in making load calculations is to group the convection and radiation effects and to take 1.65 Btu/(hr) (sq ft) (deg F) as a combined inside film coefficient, h_{cr}, for transfer from room air to interior surface. The equivalent coefficient can be defined by the equation,

$$h_{cr}(t_a - t_i) = h_c(t_a - t_i) + h_r(t_s - t_i) \quad \ldots \quad (13)$$

or

$$h_{cr} = h_c + h_r \frac{t_s - t_i}{t_a - t_i} \quad \ldots \quad (14)$$

in which h_c is the coefficient for convective transfer and h_r the equivalent coefficient for radiant transfer; the latter term is defined by the equation,

$$h_r = \frac{0.172\left[\left(\frac{T_s}{100}\right)^4 - \left(\frac{T_i}{100}\right)^4\right]}{t_s - t_r} \quad \ldots \quad (15)$$

and, for the temperatures usually found in rooms heated by radiators, convectors, or warm air systems, it can be taken as a constant having the approximate value of 1.01.

The convection coefficient, h_c, varies with position of surface, temperature, and temperature difference, but for a vertical surface under average conditions it can be taken [9] as approximately 0.7 Btu/(hr)(sq ft)(deg F). With these approximations for h_c and h_r the expression for the combined coefficient simplifies to,

$$h_{cr} = 0.7 + \frac{1.01\,(t_s - t_i)}{t_a - t_i} \quad \ldots \quad (16)$$

Equation 16 is inexact for the reasons noted, notably that both h_c and h_r have been assigned constant instead of variable values, but it does serve as a rough

[9] Radiation and Convection from Surfaces in Various Positions, by G. B. Wilkes and C. M. F. Peterson. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 513.)

expression for investigating inaccuracies in the accepted *standard* value of $h_{cr} = 1.65$.

To demonstrate the way in which h_{cr} varies, consider the room which was analyzed in the first example. For this room t_a=74.2 F and t_s=65.8, from Equation 4) so Equation 16 becomes,

$$h_{cr} = 0.7 + \frac{1.01\ (65.8 - t_{i_x})}{74.2 - t_{i_x}}$$

where the subscript x refers to a small element of exterior surface at temperature t_{ix} and having a combined film coefficient h_{crx}. This equation does not, of course, provide a true value of the h_{cr} term as it does not take into account shape factors and hence does not properly integrate the radiant exchange, but its application to elements of each different type of surface will at least show the probable range of variation. If an element of glass area is considered (by Equation 3),

$$t_{i_g} = 74.2 - 0.606 \times 1.13 \times 74.2 = 74.2 - 50.8 = 23.49$$

so,

$$h_{cr_{glass}} = 0.7 + \frac{1.01\ (65.8 - 23.4)}{74.2 - 23.4} = 0.7 + 0.84 = 1.54$$

Now considering an element of exterior wall surface,

$$t_{i_w} = 74.2 - 0.606 \times 0.25 \times 74.2 = 74.2 - 11.25 = 62.95$$

and

$$h_{cr_{wall}} = 0.7 + \frac{1.01\ (65.8 - 62.95)}{74.2 - 62.95} = 0.7 + 0.26 = 0.96$$

In spite of the multitude of assumptions and approximations which are involved in these calculations they nonetheless serve to show that a departure of over 50 per cent can occur from the accepted *standard* value of the inside film coefficient, 1.65 Btu/(hr)(sq ft)(deg F).

The intent of this paper is not in any measure to criticize the standard nor to imply that some other value or some other form should be used. Except in the case of structural materials of very small thermal resistance (such as glass or sheet iron) the influence of the inside film coefficient on the over-all coefficient is not great and deviations from the *standard* value do not therefore seriously impair the validity of the resulting over-all coefficients. The one very important point which it is desirable to bring to attention is that the *standard* film coefficient cannot be effectively used in analyses which require anything like an accurate investigation of precise surface temperatures and of rates of radiant interchange. The conclusion seems inevitable that for panel heating and cooling problems, or for problems involving the accurate determination of surface temperatures to avoid *condensation,* recourse should be had to the fundamental equations for h_c and for h_r and thence, if h_{cr} is required, to Equation 14.

The entire preceding discussion is limited to rooms in which surfaces are relatively good absorbers of radiant energy. If surfacing materials of high reflectivity (with respect to radiation of wave length in the infra-red) such as polished copper were used a further decided complication would occur and

the *standard* film coefficient would have very little meaning. This case has been given some attention in a previous report from this research project.[10]

PART III. THE OUTSIDE COMBINED FILM COEFFICIENT

In the same manner as for the inside coefficient (Equation 14) a combined outside film coefficient can be defined by the equation,

$$h_{or} = h_c + h_r \frac{t_i - t_s}{t_i - t_a} \qquad (17)$$

where t_i is now to be interpreted as the temperature of the outside surface, while t_s is now the equivalent temperature of the surround as *seen* by the outside surface. The surround temperature will be taken, as before, as a weighted average of the temperatures of all radiant receivers which are visible to the external wall. The accepted *standard* value of h_{cr} as recommended[11] is 6.0 Btu/(hr)(sq ft)(deg F) and as this is based on an assumed 15 mph wind it can be broken down to radiant and convective parts of approximately 0.8 and 5.2 respectively. In the lower temperature region of interest in design (for determination of maximum heat load) h_r decreases from 1.01 to approximately 0.6. Then Equation 17 becomes,

$$h_{or} = 5.2 + 0.6 \frac{t_i - t_s}{t_i - t_o} \qquad (18)$$

Again using the data of *Example 1,* the calculated outside surface temperatures of glass and of wall are given by,

$$\frac{t_{i_g} - t_o}{t_a - t_o} = \frac{1/6.0}{1/1.13} = \frac{1.13}{6.0}$$

or,

$$t_{i_g} = 0.188 \times 74.2 = 13.9 \text{ deg and similarly } t_w = \frac{0.25}{6.0} \times 74.2 = 3.1 \text{ F}$$

Then substituting into Equation 18,

$$h_{cr_{glass}} = 5.2 + 0.6 \frac{13.9 - t_s}{13.9}$$

and

$$h_{cr_{wall}} = 5.2 + 0.6 \frac{3.1 - t_s}{3.1}$$

Since the analysis is being carried out for conditions of maximum load, the assumption will be made that it is a clear night and that the equivalent radiative sky temperature[12] is −40 F. Likewise a first assumption will be made that 50 per cent of the radiant energy leaving the window surface and the

[10] Loc. Cit. See Note 4.
[11] HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1944, p. 91.
[12] The Calculation of Heat Transmission, by Fishenden and Saunders. H. M. Stationery Office, London, 1932, p. 37.)

wall surface is received on the earth or on earthbound surfaces; for three conditions, taking the earth temperature as 0 F, the value of t_s becomes

$$\frac{0 + (-40°)}{2} = -20 \text{ F and}$$

$$h_{er_{glass}} = 5.2 + 0.6 \times 2.44 = 5.2 + 1.5 = 6.7 \text{ Btu/(hr) (sq ft) (deg F)}$$

$$h_{er_{wall}} = 5.2 + 0.6 \times 7.5 = 5.2 + 4.5 = 9.7 \text{ Btu/(hr) (sq ft) (deg F)}$$

For a horizontal skylight which *sees* only the clear night sky,

$$h_{er_{skylight}} = 5.2 + 0.6\,\frac{13.9 + 40}{13.9} = 5.2 + 2.3 = 7.5 \text{ Btu/(hr) (sq ft) (deg F)}$$

For a flat roof with $U = 0.25$ (conventional) and $t_a = 74.2$,

$$h_{er_{roof}} = 5.2 + 0.6\,\frac{3.1 + 4.0}{3.1} = 5.2 + 8.3 = 13.5 \text{ Btu/(hr) (sq ft) (deg F)}$$

The departure of these values from the *standard* value of 6.0 Btu/(hr) (sq ft) (deg F) is sufficiently pronounced to indicate that great need exists for more careful consideration of radiation when determining outside nocturnal surface temperatures and for revising the over-all coefficient when used under conditions which give severe nocturnal exposures. In most design problems a heating plant large enough to care for the maximum load occurring during daylight hours will adequately serve its purpose. In other cases, however, need exists for meeting the maximum nocturnal as well as daytime loads.

Again attention is directed to the fact that use of the so-called standard values of either the inside or outside combined coefficient is unlikely, under usual conditions, to result in serious error. The vital fact remains, however, that deviations of 50 per cent to 100 per cent from the standard values *do* occur in normal practice and must therefore be given proper consideration if conditions are such that the films are controlling factors in determining the over-all thermal resistance.

SUMMARY

The present paper has been prepared as a progress report on the cooperative research on panel heating and cooling which is being conducted at the University of California in cooperation with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and under the auspices of the A.S.H.V.E. Technical Advisory Committee on Radiation and Comfort. Basically, however, this paper is intended not as a report on laboratory progress on the panel project but as a gathering of notes relative to the fundamentals of load determination for heating systems as influenced by radiant exchange.

Although intended as a project for obtaining and correlating data leading to the development of methods for analyzing heating and cooling systems which are primarily of the radiant type, the panel research project has necessarily been concerned with the fundamental problems of radiant exchange. These problems occur in radiator, convector, and warm air heating systems no less than in systems of the radiant or panel type. To emphasize this broader application of common principles and of common solutions to recurring

basic problems the present report has been somewhat divorced from the specialized title of the cooperative project.

Part I of this report proposes a rational procedure for determining the design inside air temperature for an enclosure heated by any conventional system of the convective type. The proposed method is adaptable to graphical solution (Fig. 1) and thereby permits realization of a simple, general, yet accurate determinaton beyond the power of existing empirical correction data.

Parts II and III are devoted to an examination of the inside and outside combined surface heat transfer coefficients for convection and for radiation. The values of combined coefficients as recommended by the A.S.H.V.E. GUIDE are not questioned when their use is restricted to their intended purpose, *viz.*, load determination for convective-type heating systems. But when surface temperatures are to be accurately determined or when radiant exchange analyses are required (as in panel heating design) then the large deviations (50 per cent to 100 per cent) which may occur between the actual combined coefficients and the *standard* combined coefficients become worthy of careful attention.

DISCUSSION

H. F. RANDOLPH, Utica, N. Y. (WRITTEN): The paper on radiation corrections shows a rational method of designing a heating system for a comfort condition rather than, as so generally is done, for a fixed temperature.

The rather laborious method of arriving at the same results in the past has deprived small structures of the benefit of proper allowance in the heat load calculation, other than from some arbitrary addition.

This simplified method, I hope, will find its way into THE GUIDE and that, for a more accurate application, a table will accompany the graph, especially for the range of over-all coefficients of heat transfer and outdoor temperatures most commonly found. I believe this paper is a splendid contribution.

H. B. NOTTAGE, East Hartford, Conn. (WRITTEN): Being privileged to speak from an interest in the physical principles dealt with in this paper, rather than from actual panel heating design experience, the subject material presents a strong appeal on the basis of the philosophy of judging design problems which it represents.

The authors deserve commendation for having pointed out the limitations which need to be considered in the too broad application of the standard average film coefficients. The future is bound to see the development of unconventional designs which will require occasional modifications from current standard procedures and it is well to keep constantly in mind the *principles* upon which such modifications will be based, in accord with the discussions of this paper.

The authors have advocated that designs which depart from established experiences in the relative balance of radiant and convective heat transfer should be analyzed in terms of the fundamental principles which govern each of these component processes. Their division of the over-all problem into its components, and the analysis of these components separately, should be an important contribution to the background of design literature being established by the Society. Generally speaking, heating design should be a true analytical procedure which has reference to both an average standard system or method and rational departures therefrom.

W. E. CROWELL, Buffalo, N. Y.: This paper is of tremendous interest to me, because just recently I had occasion to try to calculate the cooling load on an aircraft cabin. The cabin consisted of a good deal of canopy area through which solar radiation passed. I tried to solve the problem without using insulation on the skin of the aircraft, and came out with a tremendously low cabin air temperature which I

could hardly believe myself. In order to keep the men comfortable a temperature of 30 F was necessary for an air temperature.

I would like to have a comment on whether I need to revise my calculations, or whether it is expected that such a temperature would be necessary. I should add that computations were made for a wall temperature of 156 deg.

AUTHOR'S CLOSURE: In connection with Mr. Randolph's remarks, it might be interesting to note that the origin of this paper was a written discussion which the authors prepared on a paper of his. In attempting to explain some of the differences which appeared between his experimental work and the method given in THE GUIDE, we started working on the equivalent conductance idea; this led to the present paper.

As to the question in connection with aircraft, I am not able to comment. We have done, and are doing, some work in connection with aircraft radiant heating design. I will say, however, that the correction to inside air temperature based upon conditions which exist in aircraft was another of the sources of interest which led us to work on the present paper. In aircraft design at very high altitude there exists very serious need for correction. That subject Professor Raber and I have discussed in an article which appeared in the *Institute of Aeronautical Science Transactions* in July, 1944; a further discussion appearing in *Aero Digest* for December, 1944.

There are two other items that deserve attention. First, that the paper discusses an equivalent U, not an equivalent conductance. In almost all radiation work the interest is in conductance from inside surface to outside air. Second, that the curve in the present paper is not only for panel heating systems but is intended for non-radiant heating systems (warm air, etc.), as well. Note also that the curve applies equally as well to summer as to winter conditions.

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CONTROL OF INDUSTRIAL ATMOSPHERES

Dusts, Fumes, Mists, Vapors and Gases

By W. N. Witheridge,* M.S., Detroit, Mich.

MEMBERS of this Society who have their eyes on the opportunities of the immediate future, certainly have noticed with keen interest the increasing number of articles on dust control, atmospheric pollution, industrial air cleaning, airborne product recovery, atmospheric explosion hazards, health hazards and many other problems created by the existence of dusts, fumes, mists, vapors or gases in the air.

The point of interest is not that such problems exist, but that more exacting demands for the control of air quality are being made by the engineers' industrial clients. This trend will increase as the public learns more about the possibilities of improving its atmospheric environment, both within and in the neighborhood of industrial plants. It is inevitable, of course, during a period when the public mind is learning to expect higher standards of working and environmental conditions, that some ridiculously severe demands will be made upon engineering skill, and upon the resources of industrial property owners. Nevertheless, the heating and ventilating engineer is quick to realize that any widespread desire to improve the atmospheric environment during working hours is a trend in the right direction, and calls for intelligent and vigorously practical guidance.

This paper is prompted by the author's experience in observing the performance of industrial ventilating systems in a large industrial area for nearly a decade. It is a frank discussion of a few current administrative and technical problems confronting ventilating engineers who devise the equipment for controlling air contaminants produced by industrial processes.

The observations presented here have been seasoned by the personal experiences of many of the engineers and contractors serving the Detroit metropolitan area. We are greatly indebted to them for their practical advice and cooperation, and for the valuable opportunity of seeing first-hand their effective solutions of many troublesome ventilating problems.

State Laws and City Ordinances

Government agencies throughout the country are becoming more and more interested in writing codes, regulations, standards, ordinances or laws dealing with the control of airborne industrial diseases. Such codes cannot be written

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without directly affecting the work of heating and ventilating engineers who conduct their business in the city or state covered thereby.

The existing factory inspection laws of some states contain erroneous clauses or misleading provisions concerning technical engineering matters. They are details that can be amended in most cases only by the legislature—details that deserve constant review and revision in order to keep pace with developments in the heating and ventilating sciences. From a realistic viewpoint, it does not seem to be good legal administration to expect a legislature to take over the essentially continuous job of keeping ventilation engineering standards up-to-date. Responsibility for such changes might better be delegated to a recognized and non-partisan group of practical specialists who are required to open their important decisions to public hearings.

Although some of these erroneous laws are seldom enforced, they serve to create disrespect for other very desirable provisions; and even though they are not enforced, they cause a great deal of harm to the public and to the heating and ventilating profession. It is especially unfortunate when a government representative cites a faulty legal provision to justify the design of an obviously useless ventilating system. Quite possibly a provision that is too severe, by error, may force the designer to plan a much more expensive installation than necessary. If he doesn't happen to be aware of the fact that such a requirement is not enforced, he is at a financial disadvantage with respect to his competitors who ignore the law.

Many heating and ventilating engineers have had personal experience with the failure of separate official agencies to reconcile and coordinate their legal demands on equipment construction and performance, whenever their jurisdictions converge upon a given installation for atmospheric control. A workable remedy for this condition is not likely to appear in any case of such conflict until the agencies concerned are ready to agree, first, that such confusion exists; second, that it begets a regrettable waste of time and money; and third, that joint efforts of the agencies are possible. This is one of the problems that deserves the thoughtful attention of professional, non-political groups such as the A.S.H.V.E.

VENTILATION STANDARDS AND PERFORMANCE GUARANTEES

The use of performance testing instruments and air sampling devices of all types is becoming more common in industry, both as a routine check on existing equipment, and for the purpose of determining the initial performance of a new installation. They are also recognized essentials in the elaborate collection of apparatus carried around by the industrial hygiene engineer.

The ventilating engineer must be deeply interested in the standards, legal or otherwise, against which his equipment will be compared. He may have learned on several occasions that such comparisons can be made by persons who have a duty to perform, but who do not have the necessary engineering background, or the right of legal discretion to make sound modern interpretations of an old law many years out of date.

The ventilating engineer must not be too quick to guarantee that his new system will pass health or labor department tests, when such are made, unless he is fully aware of the nature of such tests, and has good evidence that his equipment will comply. Some of the standards of air quality are necessarily

quite severe. On the other hand, as soon as he is certain that he can deliver a successful product, he may include a guarantee to meet existing standards to support his confidence in good results.

In some districts, as in Detroit, he may voluntarily submit his plans to the official agency to obtain an approval or appraisal in advance of installation, or he may arrange for field tests on an experimental basis under actual operating conditions by agreement with his client, particularly if the proposal is unusual or of questionable merit. In a very few parts of the country, the agency enforcing minimum standards must approve the plans for all new or reconstructed industrial ventilating equipment prior to installation. In such cases, it is quite essential that the examining group issue clear, specific standards or instructions for the guidance of designers. The latter system cannot operate fairly unless the examining group is large enough to avoid time-consuming delays and is carefully staffed with competent engineers.

The desire of some engineers to assure their customers that health standards will be maintained by their equipment has made them acutely aware of the inadequacy or confusing nature of some of these standards. The designer takes little comfort in the knowledge that 10 cu meters of air must contain no more than 1.5 milligrams of lead dust after his new ventilating system is put into operation. Instead he needs to know what this means in terms of air cleaning load—just as he needs a generous amount of data to determine heating and cooling loads.

What the ventilating engineer should know (and this is where tremendous opportunities for research now exist) are such items as the airflow quantity in cubic feet per minute required per machine, worker, equipment area or other appropriate production unit; the *capture* velocity or controlling air velocity in linear feet per minute under many operating conditions; or the dilution factor for general ventilation (number of times an existing dust or vapor concentration must be reduced, which requires both *before* and *after* air concentration data). The maximum allowable concentration (MAC) of dust in the air recirculated to a room through an air cleaner can be of use if the air cleaner has been properly rated on this basis.

Maximum Allowable Concentration

It is the author's personal belief that engineers allied with the industrial hygiene field owe a long overdue apology to the heating and ventilating profession for their failure, so many times in the past, to state in readily digestible terms what the ventilating engineer needs or desires to know about the air contaminants he is asked to control. He is asked to reduce the concentration of X dust down to Y million particles per cubic foot of air, and then told that after he makes his first attempt, someone will return with an electrostatic precipitator and anemometer to find fault with his installation. Rarely do we bother to advise him in advance that the required air purity is nearly equal to that found outdoors on a clear day in that vicinity—or some equally revealing information.

Maximum allowable concentrations in air of substances that are dangerous to breathe must be translated into workable standards, particularly when industrial hygienists jealously maintain that air testing for extremely small amounts of dusts, fumes, or gases is a highly technical procedure. In fact, the

ventilating engineer has no intention of becoming also an analytical chemist. Furthermore, he cannot go into his fan table with a maximum allowable concentration figure, and come out with anything but a headache. This we have learned forcibly from engineers who have no time nor desire to juggle chemical equations.

The best way to get prompt, sure-footed results is to specify a ventilation rate for a given hazard, exposure, or industrial process that has a good chance of keeping the air contamination below the accepted *maximum allowable concentration.*

If this rate proves to be excessive or inadequate after repeated tests over a wide variety of operating conditions, in numerous plants under different supervisory and labor conditions, and by independent investigators, a new standard obviously is required. Development of these ventilation standards should be the joint, carefully planned projects of the A.S.H.V.E., *A.S.A.* (*American Standards Association*), *A.I.H.A.* (*American Industrial Hygiene Association*), *N.C.G.I.H.* (*National Conference of Governmental Industrial Hygienists*), and other interested professional or semi-official groups.

SEEING IS BELIEVING

The non-technical persons, the *laymen,* including both employers and employees, have really very little alternative but to judge health hazards by what they see, sense, or experience for themselves. Eventually they are in the proper frame of mind to learn by the mistakes and difficulties of others, but at the outset they learn by the most potent method available—personal experience.

This is a distinct handicap in the field of occupational health hazards, and especially troublesome when dealing with microscopic and sub-microscopic matter such as dusts, fumes, mists, vapors and gases. Why? Because many of the dangerous air contaminants make absolutely no physiological impression until the damage has been done. Workers can *carry on* for months and years under hazardous concentrations of silica dust without any discomfort or evidence of harm and finally they seem to have suddenly developed a case of disabling silicosis. They may, for instance, resent very bitterly a suggestion that they be transferred from an interesting, good paying job that creates invisible but dangerous concentrations of lead fume, because they have not yet become ill with lead poisoning. Or they may actually like the odor of the vapors of a new organic solvent which does a much better production job, and be quite unaware of the possible effects on their health of inhaling these vapors in small quantities over a long period of time. All these workers and their employers may not, as yet, have personally experienced the suffering and cost of occupational disease.

It is also a point of importance to the industrial ventilating engineer to know that the customers and users of his equipment are not always sold on the idea of having it around. Perhaps he was called in to design or install a ventilating system because a government inspector so ordered. Perhaps top management is fully aware of the dangers of a certain operation, and both the men and their supervisors find ventilating equipment already installed as soon as they arrive on their new jobs. But perhaps these men find that the type of mechanical ventilating equipment devised slows up their production

rates and they notice no serious discomfort or ill feeling when they fail to turn on the fan. Someone in the plant may eagerly promote the idea that the ventilating engineer dreamed up a useless system with a white-elephant future. The next job of ventilation in that plant goes to another designer.

The Use Factor

Why is the foregoing situation worthy of comment? Because the ventilating engineer who designs a system for the control of a certain dust, fume, mist, vapor or gas must know whether that substance is visible, irritating, offensive, or whether it has no effect whatever upon the human senses at the concentrations present in the plant air.

If his system produces an obvious and highly welcome improvement in the factory atmosphere, the users will put up with some mechanical interference on their jobs in exchange for their health or comfort. If, however, his equipment has the essentially thankless job of keeping an odorless gas or an invisible fume under control, he must take extra pains to design it with as little interference with normal plant operations as possible.

Observed many times in the Detroit area has been the scrapping of expensive process ventilating equipment that failed to take serious account of (to use the author's own terminology) *the silent sabotage of a very low Use Factor.* It might be a mechanically ingenious design, built to endure, pleasing in appearance and efficient in operation, but such a nuisance to operate in a mass production plant that its *Use Factor* is strictly zero.

Common Causes of Failure

Experiences with investigations in Detroit in over 2500 industrial plants have revealed all of the following causes of failure or waste in ventilating equipment for the control of industrial atmospheres. These difficulties are a matter of record, and are not simply a tabulation of potential faults.

1. Inadequate air supply for an exhaust system creating an air-bound room. This is chiefly a winter problem in the Detroit climate and often occurs when a system is installed in summer with plenty of open windows and doors, and no check is made of its performance under simulated winter conditions before the installation is accepted.

2. Short-circuiting of the ventilating fan or hood by the effect of an open window, door, skylight or stack near it, a condition which might be called *ventilating the outdoors.* The frequency of this fault is amazing.

3. Locating ventilated equipment in the vicinity of open windows permitting entrance of powerful drafts of air which impose an almost impossible task upon the ventilation. This is very common with solvent type degreasers and other open-top ventilated tanks.

4. Exhaust hood located too far from source of air contaminant. It seems very difficult for plant operators and workers to appreciate the rapid deterioration of air velocity with distance from an exhaust opening. The fact that air moves from all directions toward an exhaust opening seems to be *one of the wonders of the world.*

5. Use of general ventilation with resultant waste of heat, when a local exhaust system handling a small amount of air would be practicable.

6. Backward operation of a centrifugal fan. Many persons in industry are not aware of the fact that a centrifugal fan, as opposed to an axial flow fan, will move

air in only one direction (from the center outward) whether it runs forward or backward. Its capacity in reverse is, of course, greatly reduced.

7. Excessively high duct velocities when transport of particulate matter is of no concern. Conversely, insufficient transport velocities in ducts carrying dust. The accumulation of flammable dust is especially dangerous.

8. Sheet metal construction that ignores present knowledge of low resistance design.

9. Substantial underestimation of the pressure losses of an exhaust system. This occurs often with flexible-type ducts.

10. Construction of ducts and hoods that are practically impossible to clean and failure to use convenient cleanouts when provided.

11. Failure to protect all parts of the ventilating system against rapid destruction by corrosive atmospheres.

12. Selection of fans by *free-flow* ratings. Even axial flow fans as used in industry for general ventilation seldom operate under free-flow conditions.

13. Excessive leakage of dusts or gases from positive pressure ducts that could readily be operated under suction to reverse the direction of leakage. Industrial equipment is bound to develop leaks continually.

14. Failure to provide enclosures, baffles, or windbreakers whenever possible to reduce the quantity of air to be moved.

15. Overloading an existing system by adding exhaust openings not contemplated in the original design.

16. Design of a system to create a specified static pressure instead of the minimum successful rate of airflow.

17. Complicated designs of local exhaust units that are too difficult to use.

18. Drawing dusts, fumes or gases toward, instead of away from the worker's breathing zone.

19. Blowing unheated or untempered make-up air at workers during the winter season.

20. Recontaminating plant air by discharging contaminant outdoors at a point where it can easily return to the plant.

21. Recirculating contaminated air from an exhaust system back to the factory atmosphere without effective cleaning.

22. Use of the wrong type or size of air cleaner. This includes use of too small or too large cyclones, insufficient filtering area, wet collection of dusts that resist wetting, single stage collection on a very heavy dust load with a great range of particle sizes, and (quite unbelievable!) ozone generators still being installed for the purpose of eliminating carbon monoxide!

A complete listing of similar causes of failure observed to date includes about 50 items.

SUB-STANDARD INSTALLATIONS

One of the bread-and-butter problems confronting the conscientious and competent ventilating engineer is the loss of an order to a concern that specifies equipment that he knows has little chance of complying fully with health or labor department requirements. Perhaps it is known that enforcement of industrial health standards is inadequate or very tardy in a certain city or state and therefore the industrial customer takes a chance on a less costly or makeshift installation which he hopes will pass government inspection. Possibly the customer may gamble on the fact that the installation will never

be given more than a casual, visual, or hasty visit by an inspector eager to credit him with a compliance with his official order. Occasionally the designer takes the attitude that the government-sponsored standard is sky-high and nonsense. (Unfortunately this is true of some laws as was pointed out earlier.)

Ventilating engineers who encounter this situation throughout the country are anxious to see better enforcement of minimum standards—those which they personally know to be quite necessary if honest value is to be given for the customer's ventilation dollar.

ALL-PURPOSE VENTILATING EQUIPMENT

The author and his associates have observed an almost monotonous tendency on the part of many plant operators to approach an air handling problem as if it were quite as elementary as the opening of a bedroom window. They are easy prey to the ill-advised person who promotes his product as the universal solution for all problems of factory ventilation. They are the customers who are easily misled and then become disgusted when mention is made of *ventilation* or *ventilating engineers.*

All-purpose equipment is offered by the *expert* who always recommends a fan in the wall and may remind the customer that he has done so for years and years. Another will promote a dust collector that works on everything, including vapors and gases. There is the natural draft enthusiast who always finds a dependable wind around the corner. There is the canopy hood expert who puts a hood *over* every problem and expects the air to be self-propelling in an exclusively upward direction; and the small-slot, high-velocity, low-volume specialist who seems to be thinking only in terms of his household vacuum cleaner. All such men perhaps are victims of the one-track mind rather than possessors of unethical tendencies. Still they manage to find enough customers who want something for practically nothing, and, in the end, much of their work has to be undone before a genuine health hazard can be brought under control. As a matter of fact, their existence is one of the reasons why the A.S.H.V.E. is necessary to protect the reputation and interests of the heating and ventilating profession.

COMBINED SAFETY AND HEALTH PROBLEMS

There has always seemed to be greater respect for explosion hazards carried by industrial atmospheres than for hazards to the health of workers breathing the same air. In the usual manner, knowledge of such possibilities for accident has been obtained by suffering many tragic and costly industrial explosions that are immediately and drastically impressive. At present, it is much easier to get an industrial executive to eliminate the explosion hazard in the handling of a volatile liquid than it is to make him concerned about the slow and quiet development of an occupational disease to which the workers might also be exposed. In fact, if the explosion hazard is controlled, he is quite anxious to consider the health hazard equally under control.

Unfortunately, it so happens that the concentrations of flammable vapors that are kept to ¼ or ⅕ of the lower explosive limit to provide a reasonable safety

factor, are still not so low as they must be if the vapor-laden atmosphere is breathed by human beings. There are very few exceptions to this statement. The lower explosive limits of most vapors are above 1 per cent by volume (10,000 parts per million). If ⅕ of the lower explosive limit is permitted, 0.2 per cent or 2000 ppm would be a safe concentration for such vapors from the fire standpoint. From the health standpoint, the permissible concentration for industrially common vapors ranges chiefly between 25 and 500 ppm, with very few that can be tolerated above 1000 ppm. This is one of the lessons that the ventilation-conscious safety engineer must learn if he has the responsibility for, or interest in, the health as well as safety of his workers.

RECIRCULATION OF FACTORY AIR

The current necessity for fuel conservation has brought into sharp relief the demands for permission to recirculate to the general factory atmosphere the air used to remove excessive process dusts or other air contaminants. Some of the most important recent advances in industrial ventilation technique have occurred in this direction. Such installations impose the most severe demands upon industrial air cleaners of all types when the air they discharge must be clean enough to breathe all day without harm to health. Many air contaminants of a highly dangerous nature create atmospheres that defy the best commercially practical air cleaners yet devised and, consequently, must be discharged outdoors through high stacks along with valuable heat units. Collectors with a stated 99 per cent efficiency may still pass enough of some of these dangerous contaminants to make the air unfit for breathing over long periods of time. In general, the hazardous gases, vapors and fumes are most difficult to remove while the dusts and mists offer the best opportunities for recirculating systems. Most manufacturers of air cleaners are well aware of the limitations of their equipment and are anxious that their products will not be subjected to hopelessly severe operating demands.

NEIGHBORHOOD NUISANCES

The Bureau of Industrial Hygiene in Detroit, as is also true of similar investigating agencies in other communities, is keenly aware of the perennial conflict between industrial buildings and their commercial and residential neighbors for relief from allegedly offensive, sickening, irritating or destructive air contaminants discharged by a ventilating system designed for the benefit of plant occupants. In fact, as time goes on, conditions inside factories are becoming better and better and surrounding atmospheres are accordingly more and more contaminated. This indeed presents a worthwhile problem to the designer and manufacturer of air cleaning equipment for the collection of dusts, fumes, mists, vapors or gases.

ANTICIPATING NEW CONTROL PROBLEMS

The new era of synthetic organic chemicals developed to replace the shortages in our material resources already has multiplied many times the instances where buildings and their mechanical equipment become unable to cope with man-made chemical-bearing atmospheres. It is not easy for the designer to

keep in step with the constant changes in the materials of industrial production. For example, the heating and ventilating engineer may have installed a very desirable modern industrial air conditioning plant which quite unintentionally picks up the vapors of a new offensive chemical and distributes them to every corner of the plant. In July the dust collecting engineer may install a new emergency exhaust system that completely overwhelms the heating plant in December. The architectural engineer may design a new industrial building that soon must be partially torn apart to accommodate air control equipment which, as the ventilating engineer might claim, in retrospect, was obviously necessary.

Although ventilating engineers certainly cannot be held responsible for production changes that affect plant atmospheres, those who can advise and warn their clients or employers about the air conditioning aspects of new processes, are today in a most enviable position. In fact, if the ventilating engineer simply relays the information he has obtained from known industrial hygiene sources in his community, he is credited with knowing the latest developments in his own field.

THE INDUSTRIAL HYGIENE ENGINEER

The industrial hygiene profession is sometimes loosely divided into two major groups: medical and engineering. To those who design, buy and sell industrial ventilation, this fact is important. Ventilation is perhaps the most useful exposure control method at the disposal of the industrial hygiene engineer. From 50 to 75 per cent of all occupational disease hazards encountered in his daily work (excluding skin diseases) terminate with the requirement of some form of ventilation.

In some organizations an industrial hygiene *engineer* is not necessarily an engineer, and seldom has he entered the field as a ventilation engineer. He may be basically a sanitary engineer, civil engineer, chemical engineer, chemist, biochemist, physicist, electrical engineer, mechanical engineer, ventilating engineer, or any other technically trained person who has a valuable background to offer the field of industrial hygiene, and who would not be termed a medical industrial hygienist. Industrial hygiene needs all this variously specialized personnel to carry out an effective study of industrial health problems and to arrive at successful methods of control.

Thus, technical ventilating problems daily confront all types of industrial hygiene engineers who may be called upon to approve or disapprove existing or proposed methods of controlling an airborne hazard. They may be required to do this as government officials responsible for safe working conditions under their jurisdiction. As a group, therefore, industrial hygiene engineers try to learn as much as they can about the possibilities and limitations of control by ventilation so as to avoid costly errors of judgment and counsel.

At the same time it is desirable for mechanical engineers who specialize in air conditioning to learn, during their university training, of the ventilation problems related to industrial hygiene. Until very recently it was quite natural, because of limited course opportunities, that many students passed over the possibilities of this specialty as not even being within the realm of the engineer. It is gratifying to note that more engineering schools are beginning to carry this subject in their heating and ventilating curriculums.

COLLABORATION WITH THE A.S.H.V.E.

The Society represents the resources and talents of hundreds of engineers who have learned to handle the *thin air* with confidence and respect.

On the other hand, there are so few ventilating engineers available for full-time industrial hygiene work that many of the engineering investigations and studies on air handling problems associated with industrial health hazards are carried out by industrial hygiene engineers in various parts of the country who admittedly have only a casual background in the fundamentals of ventilation engineering. This statement must not be misunderstood. In a sense, these efforts of members of other professions to solve our problems have been most healthy and valuable additions to the information at our command. Nevertheless, it is believed that the most reliable ventilation engineering research can be conducted by or with engineers who have devoted their years of experience to this field.

It is not wise for any technical group to undertake the study of special problems in industrial ventilation without tapping and exhausting the existing resources of the A.S.H.V.E. One of the serious difficulties we all experience in following the rapid development of industrial hygiene engineering techniques, is the very much scattered and uncoordinated state of our present efforts in ventilation research.

There are many industrial hygienists who are quite unaware of the excellent and potential services afforded by the Society's Research Laboratory, coordinated with other research and educational institutions. Conversely, it is believed that a large number of the Society's members do not know of the nation-wide and local influences that industrial hygienists are exerting on their current heating and ventilating activities.

The author firmly believes that there exists an urgent need for more intimate relationship between the A.S.H.V.E. and those public health and industrial hygiene engineers who use ventilation as a major weapon in the prevention of occupational disease.

DISCUSSION

L. T. AVERY, Cleveland, Ohio. (WRITTEN): We are indebted to the author for daring to criticize the heating and ventilating technique that has been so common in the industrial plant. Handling air is not easy, and to do a good job of industrial air conditioning requires topflight engineering, and plenty of money. Too many gadgets have been sold as a temporary palliative to the plant manager who grabs at a low cost, quickly secured fan, which visibly and audibly handles a *lot of air* but which quiets the complaint only while the item is on order. When finally the installation is completed, the season has changed, or maybe the complainant was fired or promoted and another year rolls around with nothing really accomplished.

The author touches the whole field of industrial ventilation, but I wish to comment on one in particular that is right now at the top of the list, and that is ventilation of foundries. Grey iron castings are a bottleneck in the war program and new emphasis is being put on *cleaning up the foundry.* Apparently foundries have been neglected by the heating engineer because there was so much wild heat that it seemed like *carrying coals to Newcastle* to install a decent heating system. There was so much smoke that no one really dared try to collect it, and silicosis, pneumonia, and tuberculosis were considered as a functional weakness of the particular type of worker

rather than direct results of the quality of the air. When you can't hire enough workers to keep your plant busy you begin to wonder why. If the absence rate is 500 per cent greater in your foundry than in the adjacent machine shop you ask questions. If the industrial commission sets up a merit rating for your foundry and you get charged with your own silicosis cases, then maybe you decide it is not an expense, but a good investment to provide a good working environment, airwise.

The Ohio Industrial Commission states in its December, 1944, bulletin, "*Silicosis originating in foundries continues to be one of the major problems confronting the Industrial Commission of Ohio.*" A recommended schedule of sanitation is established, including among other things a suggestion of 15 changes of air per hour in a mechanical foundry. That is a lot of air! And in zero weather that represents a lot of heat. And unless you clean it that includes a lot of dirt to throw over the fence at the neighbors. One of our industrial plants was recently given permission and priority to remodel its office into a windowless air conditioned enclosure because the neighboring foundry spewed forth such quantities of dirt and smoke that open window ventilation was *detrimental to the health of the workers.* Mind you, the health of the workers! Nothing was done about the health of the foundry employees working continuously *inside the foundry.*

Major James Roeder Bell, Cleveland, now medical officer in the Air Corps, used to say that the phrase *air conditioning for comfort* was a misnomer. He said the body at comfort was the healthy body and all a doctor could do was to relieve the symptoms of discomfort, whereupon the patient became well. Air conditioning has come to be identified with summer cooling, and summer cooling is considered for comfort only, and therefore on the shelf for the duration. But in the broader, and more accurate sense, air conditioning in the foundry must be the correct term to use. How else do you cover the entire problem of collecting the dirt at the shakeouts and knockouts, removing it from the air and discharging it at a quality and location that does not menace your neighbors or contaminate your own adjacent building? You have the smoke and oil vapors and gases to remove, without drawing them through the worker's breathing zone. What of the heat, the terrific, incessant, penetrating heat that opens up the capillaries, overstrains the heart, and dehydrates the worker? After you plan the exhaust systems and equipment you must provide equivalent supply air, and this must be good air, not recirculated smoke and dirt. In the winter the supply air must be heated and distributed to promote the health of the worker. In summer that air must be as cool as possible, not taken off a hot roof, and it must be distributed to flow through the workers to the heat of castings and cupolas, instead of vice-versa. Pedestal recirculating fans are a menace and create more problems than they solve, often blowing smoke and heat away from one worker directly onto another. He who *hollers* loudest wins. The successful heating and ventilating of the foundry, the *cleaning up of the foundry* if you like, is the highest type of air conditioning and unless the problem is respected it will not be solved.

Many of you recall the old argument championed by Dr. E. Vernon Hill about outside air *vs.* fresh air. He said there was no such thing as fresh air! That is particularly true around a foundry. So the next step will be to cleanse and purify the supply air, which can be done easily with present-day equipment, so that recirculated air will be purer—or fresher—than outside air. Then the excessive heating loads will be brought within reason, and the foundry that now uses a 2500 hp boiler for heating 1,750,000 cfm will be heated by a 200 to 300 hp boiler when the foundry is running light and the wild heat will do all the heating when the production is under full blast. Codes now prohibit the recirculation of exhaust air back to the foundry, even though cleaned, but the fellows who wrote those codes did not know what clean air was. Certainly nothing is said about where the new air comes from and in many cases it actually does short-circuit from the exhaust. What a glorious field is opened to the air conditioning engineer, who really can cooperate

with the industrial hygienist, with management, with safety and personnel departments, and with the industrial commissions and insurance companies to provide a healthy and comfortable working atmosphere, *even in the foundry!*

A. D. BRANDT, Chicago, Ill. (WRITTEN): My first reaction to this paper was merely to say *good work—I heartily endorse it.* However, since emphasis is gained by repetition, it may be well to add further emphasis to several of the issues which, in my opinion, are very important.

My experience in a variety of plants, both large and small, scattered throughout the country, checks that of the author's as regards the much-too-large percentage of ventilating systems for industrial atmospheric sanitation which are wholly inadequate; very much over-designed; or interfere with operations to an excessive extent. This is *not* a condemnation of the industrial engineering personnel at these plants. These engineers are busy men—they have a thousand and one things to do of which the design of a ventilating system to control dust, fumes, or gases is probably, in their estimation, very minor and something of a nuisance. Atmospheric sanitation is such a small part of their work that they cannot be specialists in the field.

Furthermore, while proper ventilation is the most important single method of accomplishing industrial atmospheric sanitation, in most instances the proper control of a hazard comprises the use of a combination of control measures including proper ventilation, but not ventilation alone. An adequate, efficient, and inexpensive design job can be done only by someone who (1) appreciates the full significance of the hazard; (2) has a good knowledge of the physical characteristics of atmospheric contaminants; (3) has had experience in atmospheric sanitation and recognizes the relative value of the several measures of dust, fume, and gas control; (4) has sufficient knowledge of industrial toxicology to decide which control measures are best fitted for the operation under consideration; (5) has a working knowledge of the fundamentals of air flow into various types of suction openings; and (6) can design a balanced exhaust ventilation system. This certainly does not look like a good summary of the qualifications of a ventilating engineer and should serve to demonstrate that industrial atmospheric sanitation or industrial hygiene engineering is a specialized field. It is a field for which sanitary or chemical engineers with special training in industrial toxicology and ventilating engineering, or ventilating engineers with special training in chemistry and industrial toxicology, are best fitted.

Thus it should be apparent that industrial hygiene engineering is not an *offspring* of ventilating engineering, even though proper ventilation is the most important tool of the industrial hygiene engineer. The ventilation employed by the industrial hygiene engineer is, by and large, a special type of ventilation and in many instances, if the system is designed properly, produces little ventilation as defined by the A.S.H.V.E. In fact, the more efficient the system the less ventilation is actually produced. Local exhaust ventilating systems in most instances are poor excuses for ventilating systems in the true meaning of the word—they are pneumatic collecting and conveying systems designed to capture and transport as much dust, fume, or gas as possible with as little air as possible. Frequently they do not even figure importantly in the heating and ventilation of buildings. Nevertheless, most of the fundamental principles involved are those of ventilating engineering and consequently industrial hygiene engineering should, and eventually must, be of direct interest to the A.S.H.V.E. It seems to me that this co-relationship, this interdependability, between ventilating engineers and industrial hygiene engineers, cannot be ignored much longer. The time has come, I am confident, when it is imperative that the A.S.H.V.E. establish a committee which will cover all phases of industrial hygiene engineering, not only one or two phases such as *Air Cleaning* and *Air Conditioning in Industry*.

W. L. BRYAN, JR., Long Island City, N. Y. (WRITTEN): This paper is most timely. The increased stress of war production has broadened the field of industrial hygiene and brought its advancements and needs into focus. In that section of indus-

trial hygiene which deals with environmental control, this paper gives a most enlightening and thought provoking analysis.

Those of you who have participated to any degree in this field, I am sure, will go along with me in corroborating the experienced information presented by the author. Mr. Witheridge hit a very important *nail on the head* when he stated the need for the *ventilation rates required* to control the hazard of a given industrial process.

In the industrial hygiene field we have a large amount of information in the maximum allowable concentrations. On the other hand, in the field of ventilation, we have a large amount of information on producing ventilation at a particular rate and quantity over an area. What we need now is to combine these two factors. Government agencies in writing codes have attempted in certain cases to do this. The results are none too satisfactory.

It has been my feeling that the amount of ventilation required to reduce a given contamination below a certain allowable concentration is a problem for the ventilating engineer. As such it is of immediate concern to the A.S.H.V.E.

P. W. GUMAER, New York, N. Y. (WRITTEN): We are fortunate in having such an excellent paper on industrial ventilation presented by one with such long and varied experience.

Mr. Witheridge has done an excellent job in the preparation of the list, *Common Causes of Failure.* His remarks on maximum allowable concentrations, however, favor an over-simplification which is not warranted if maximum efficiency and economy are to be attained.

In 1930 I made the following statement at the 19th Annual Safety Congress:

"The ventilating engineer talks in terms of temperature, humidity, velocity of air, or quantity of air removed. These are his standards of ventilation. They are sufficient in dealing with problems of ordinary ventilation such as theaters, schools, hotels, etc. These standards of ventilation are of little value in obtaining effective ventilation of solvent vapors, especially when they are heavier than air. The proper standard of ventilation of industrial vapors is the amount of the vapor remaining in the air that the workmen breathe. That is what we need to measure. The other standards may be useful in obtaining the desired result. Unless we have a means of measuring the amount of vapor breathed by the workmen, we are never sure of the effectiveness of ventilation equipment."

Industrial hygienists today appreciate the necessity of determining the concentrations of toxic materials in workroom air. The ventilating engineer, if he is not to be outdone by his competitor, must understand maximum allowable concentrations. He need not be qualified to make actual determinations but he should understand the accuracy and limitations of the methods in general use.

The air conditioning engineer who is unfamiliar with the chapter on Physiological Principles, in the HEATING, VENTILATING AND AIR CONDITIONING GUIDE and who goes into his fan table with a comfort zone figure would also come out with a headache.

In this connection may I suggest that statements on maximum allowable concentrations in THE 1944 GUIDE be revised. No reference is made to the official figures adopted by the *American Standards Association.* A definition of maximum allowable concentration and an explanation of its use should be given.

PHILIP DRINKER, Cambridge, Mass.: I am strongly in sympathy with some of the author's criticisms. I have been a member of this Society for many years, and I have yet to see any indication of general interest by the membership in industrial ventilation problems, especially in the matter of dust and fume control and poisonous gases.

Frankly, I think we miss a good bet. In my own consulting work I have to handle a good many such things for mining and manufacturing concerns and almost never does the ultimate installation go to a contractor who is a member of this group. There are a few very definite exceptions to that, and I know firms now who are in air filtration work of various kinds, who do stray over into industrial problems, such as mining and manufacturing firms offer, but, by and large, our men do not get the jobs. I have often wondered why. Usually the jobs are perfectly clear-cut. The ones

that I have seen have often run into considerable sums of money. They are for substantial organizations and there is no question of not being paid for your work. You are paid. I frankly think that our members are missing a good thing.

Furthermore, your help is actually needed. Firms that get into trouble from poor dust, fume, and gas control are very often those who have had a tin knocker's job thrust upon them to satisfy some complaint that conditions are wrong. When equipment bids are received the buyer of the job does not know very much about it, and is prone to take the low bid and thus ends up with a poor job which has to be done all over again.

Really, I think that our membership, with the knowledge gained over years of work on air conditioning jobs, could get into the industrial field a good deal more than they have. I hope they will in the future.

C. H. PESTERFIELD, East Lansing, Mich.: This paper points out many of our shortcomings in the field of industrial exhaust ventilation, as well as general ventilation intended to be used for the control of atmospheric contaminants. There are many points that need to be clarified. It is difficult to obtain authoritative data from the medical profession on the rate at which the human body can accept harmful constituents in air, but, I believe, that doctors usually are more conservative than many designers of equipment for control of the contaminants.

This paper also presents a direct challenge to the Society to establish acceptable standards for air velocities, duct sizes, air quantities, hood design characteristics and capture velocities which could be used by all engineers. If the Society could bring together various groups and societies interested in industrial ventilation it would result in an outstanding service to the public and the engineering profession.

H. M. NOBIS, East Cleveland, Ohio: This paper is a timely and valuable contribution for the practical use of the heating and ventilating engineer employed in industrial plants.

The physical detriments caused by many plant processes bring about mental irritation and energy sapping environments which, in turn, create department dis-harmony. A humanistic heating and ventilating engineer will detect unhealthy conditions before complaints arise and prevent, in many instances, minor problems from reaching mountain proportions.

Applying the *Golden Rule* before it comes to the damning stage makes for department harmony in these problems.

Many headaches could be prevented, if the production departments would acquaint the heating and ventilating engineer with the nuisance creating processes, before the equipment order and installation areas have been jointly discussed.

C. S. LEOPOLD, Philadelphia, Pa.: I agree with Mr. Gumaer's criticism of the chapter on Physiological Principles in THE GUIDE as regards contaminants. THE GUIDE publication is my responsibility for the moment. Mr. Gumaer is invited to send me his available data on the subject.

This is not a small request. A great many of the constants are not known and it is frequently necessary to search German, French, Japanese and Russian literature as, in many cases, these countries enacted regulations prior to enactment in this country. The discrepancy in data is sometimes of the magnitude of 15 to 1.

J. M. KANE, Louisville, Ky.: There is just one comment that I would like to add in view of the very apparent interest evidenced here. There is a need in the United States for a recognized national group qualified to recommend standards for industrial exhaust ventilation.

At least six states have already done excellent, outstanding work in producing codes covering control requirements for certain applications. Unfortunately, as the author pointed out, the codes are not always in agreement. A dust that can be conveyed in one state with 4500 fpm may require 6000 fpm across the border.

I believe that all states will institute more and more studies and regulations in the next few years. If a nationally recognized group with an engineering background,

such as the A.S.H.V.E., were in a position to act in an advisory capacity, a great number of conflicting requirements might be prevented in new state regulations covering the control of industrial atmospheres.

C.-E. A. WINSLOW, New Haven, Conn. It is an admirable thing that this subject has been brought before us. There is a great field here for the ventilating engineer, and I concur entirely with what has been said about the difficult situations that now exist.

I would like to point out, however, that there is a very good reason why many ventilating engineers have fought shy of this subject, because it is an extraordinarily difficult one. You cannot find an answer in the book. Each process in relation to the next process in the shop creates a special condition. Furthermore, conditions constantly change. You design a system of this kind, and then the process in the shop changes and the thing goes wrong. It is an extraordinarily difficult situation. But there is a tremendous field here, and it is very important that the engineers and the public health authorities be brought together in solving the problem.

The author did not mention the industrial hygiene section of the *American Public Health Association,* as a body from which he wants cooperation. You have to realize here that these directions, these specific rules and standards are promulgated and are going to be promulgated, in most cases, by the health departments or, in certain cases, the labor departments, of 48 states. Reasonable uniformity will have to come through the *American Public Health Association* and the U. S. Public Health Service, because the people who frame these regulations know little about the *American Standards Association.* It is primarily the health authorities who are going to formulate these standards.

We have to work with them in the development of the fundamental standards to some extent, although that primarily is a medical question, but certainly with regard to methods of meeting the medical standards.

I am sorry we are not in better position to take our part in this picture. I have been extremely interested for five or six years in our technical committee on the removal of dusts and fumes, and it has not progressed very far. The Council of the Society has been pushing hard on it, and I hope and believe that enough fire has been built under it so that we will get action. I think that is the organ of the Society through which this sort of thing should be focussed. It is a vital problem and there is a great future for the engineer in this field.

L. G. MILLER, East Lansing, Mich.: My only object in appearing here, following Dr. Winslow, is just to invite you to constitute yourself a committee of one to heckle the President until he sees that the Council and the Committee on Research collect all the folks who are interested in this field and gathers enough material so that THE GUIDE can be an appreciable textbook on that subject. If you are inclined to say, *well, let's wait until after the war,* here is one little bit of information that some of you may not be thinking about, and it may not be as bad in your case, but in Michigan our absenteeism is in the nature of nine per cent—nine per cent of the total time that is lost is due to absenteeism. I personally believe that a very large percentage of that time is lost because the working conditions are not healthful.

W. L. FLEISHER, New York, N. Y.: The Technical Advisory Committee on Air Conditioning in Industry, which is extremely interested in the subject, has been asked to distribute its activities and to form a new committee concerned only with exhaust ventilation. Rather than do that, the Director of Research has instituted an investigation of the codes of all the states applying to both exhaust ventilation and to make-up air. This is being carried on, I think, more actively than any other research project at the present time.

Our very small personnel at Cleveland has been trying to coordinate the codes of 48 states and find that in many cases a state may have a code requiring a certain amount of exhaust ventilation or a certain amount of fresh air, while the code

of a city within that state may require different amounts. For instance, in New York State the code is entirely different from the code in New York City.

We had a very definite feeling that until we found out what had been done by the different states and cities it was unwise to appoint other committees or to divide existing committees to tackle this particular subject. The one thing that has shown up very definitely, and is now being called to the attention of several of the state legislatures, is the fact that, although exhaust ventilation is required in many cases, it is inoperative because there is no make-up air provided to take care of the exhaust. Consequently, we are trying to coordinate this work. You can not make codes and you can not have laws until you know what you want to do. This work is being very actively carried on by the Committee on Research in Cleveland at the present time. Mr. Tasker said that this was the most active work of the Committee on Research at the present time. Mr. Avery was one of the people who wanted to divide the Technical Advisory Committee on Air Conditioning in Industry into an Exhaust Committee and a Supply Committee, and we felt very definitely that the whole thing should be considered as a whole and as one of the most important projects that the Society has before it.

C. H. Flink, New York, N. Y.: This is a good time to mention that the Society has been concerned with at least 75 different standards and codes. The Society has been active in connection with many types of codes and standards, some of which have a direct effect upon public health.

Up to the present time we have completed histories of about 75 codes for the use of future committees, which may be assigned the task of revising former codes or preparing new ones. The Society has a right to be proud of the record made in the past in establishing standards and codes. You will be pleased to know that there is greatly increased activity in connection with codes at the present time.

Author's Closure: I agree with Dr. Winslow that the *American Public Health Association* must collaborate with our Society on this problem. Likewise, in addition to the organizations named in the paper, we should have the assistance of the U. S. Public Health Service, U. S. Department of Labor, National Safety Council, *National Association of Manufacturers, American Society of Mechanical Engineers, American Chemical Society,* and no doubt others. Furthermore, this is a technical problem that has no business being kicked around in the manner of a political football, as appears to be the case in a few states. The best assurance of a non-partisan, ethical engineering development in the field of industrial ventilation for preservation of health is, in my opinion, the immediate sponsorship by the Society of a research program coordinated by a Technical Advisory Committee on Industrial Process Ventilation.

I am gratified by the evident interest in the subject of this paper. I sincerely hope that efforts will be made to perpetuate this interest.

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No. 1276

MINE VENTILATION AND ITS RELATION TO HEALTH AND SAFETY †

By D. Harrington,* Washington, D. C.

VENTILATION is the keystone of safety in underground *coal* mining and of health in underground *metal* mining. Effective ventilation of underground workings requires the establishment and maintenance of such control of air that underground employees may work in safety, with maximum comfort and efficiency and without impairment of health; that the mine openings be made subject to such flow of air as to remove from the workings at ordinary times harmful gases and dusts; and that at times of emergency, such as fire or explosion, as much or as little air as desired may be circulated to any or all parts of the mine.

Control of air flow is absolutely necessary and is obtainable only by installing mechanically operated fans and other ventilating devices, such as doors, overcasts, regulators, etc. From the outset, every underground mine should be equipped with a fan. There are few, if any, coal or metal mines where natural ventilation, even in ordinary times, supplies safe or healthful conditions for underground workers; and when a fire or explosion occurs, mines that depend on natural ventilation are virtually helpless and certainly are dangerous to those unfortunates forced to be in them.

Although ventilation and its control have almost always been deemed integral parts of coal mining, little attention has been paid to air circulation in metal mines until some untoward condition or accident has made it necessary, yet the need of efficient circulation of air is at least as necessary in metal mines as in coal mines. The dangerous, explosive gas, methane, as well as fumes from explosives, and in some places other gases, such as CO_2 or hydrogen sulphide or nitrogen, must be removed from coal mines. In metal mines there is much greater necessity of removing fumes from explosives because much larger unit quantities are used and frequently, CO_2, nitrogen, occasionally methane and other gases, must be removed from strata. Circulating air currents are urgently needed to reduce the excessively high humidity and temperatures found frequently in metal mines but more seldom in coal mines. The immense quantities of minutely fine particles of rock dust that float in the stagnant air of metal mines and are very largely responsible for miners' consumption and other diseases, so prevalent among metal miners in many regions, could be removed very largely by adequate ventilation. The general conception prevails that coal miners have a relatively healthful occupation while many metal miners contract diseases such as lead poisoning and miners' consumption and either die early in life or are incapacitated in middle age. The

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difference in effect upon health is due almost wholly to the superior working conditions frequently found in coal mines—chiefly to better ventilation.

Larger metal mines are comparable in many respects to an immense office building or hotel. The various levels correspond to the floors of the building, except that in a mine they are 100 ft or more apart vertically instead of 10 or 20 ft as in a building. The drifts and crosscuts correspond to the halls and corridors and the raises, stopes, and other working faces to the offices, sleeping rooms, etc. In the mine, however, there are no openings to correspond to windows to the outside, and, too frequently, no openings between working places to correspond to interior doors between offices or rooms. In many cases this leaves only one opening to the working place. In general, the present practice, even in the better-ventilated metal mines, is to cause currents of air to flow along the main drifts and crosscuts with only such volumes of air going into the working places as might seep by diffusion through one opening. The workers usually are engaged in some fairly strenuous operation such as drilling, timbering, or shoveling, which tends to fill the air with fine dust (the most dangerous kind); or they may blast to bring down rock or ore and thereby liberate not only clouds of fine dust but also smoke laden with poisonous gases. Too frequently the surrounding walls are damp and at a temperature of 80 to 100 F and the stagnant air may be at a temperature of 80 to 100 F or even higher, while the relative humidity may be 90 to 100 per cent.

Coal mines usually are required by law to have a ventilation system, but in many mines compliance with the provisions of the law is anything but complete, and as a consequence accumulations of explosive gas ignite in scores of coal mines every year and cause a heavy loss of life.

Approximately, in order of their importance, the main features that affect air in metal mines are (1) air movement, (2) temperature, (3) relative humidity, (4) gases, and (5) dusts. In coal mines the order of importance would probably be (1) air movement, (2) gases, (3) dusts, (4) relative humidity, and (5) temperature.

The matter of movement of air from the surface, through the working faces or places, and then back to the surface is by all means the most important consideration in effecting adequate ventilation of mines, and in protecting the health and promoting the efficiency of the underground workers. If adequate quantities of pure air are directed to and through the places where men work, most of the requirements of health and many of the demands of safety and efficiency will be well served.

The temperature of underground air is affected by temperature of outside air in varying degrees, depending upon the depth and extent of the workings, upon air velocities, and other considerations. The temperature of mine air is very definitely affected by underground rock and water temperatures, by the quantity of air flowing, by oxidation (or decay) of timbers, coal, and ores, and by mine fires. It is also affected to some extent by friction, moving of ground, firing of shots, and heat from lights and from breathing of animals. Heated air obtained from other mines and from contact with electric motors and other machinery may seriously affect the local temperature of air of underground workings.

Relative humidity of underground air is affected to some extent by humidity of surface air, but much more vitally by moisture of surfaces of underground

workings and especially by water dripping through the air. Quantity, temperature, and velocity of air flowing also ultimately affect the humidity of underground air and influence the effect of the humidity on the workers. Water used in drilling, wetting ore, coal or mine surfaces, or in sprays, water blasts, etc., also contributes to mine-air humidity.

Gases in mine air may come from surface air, from breathing of men and animals, from lights used, from firing of explosives, from compressed air used with machines or in blowing, from operation of various kinds of machinery including fans, and from various coal, rock, or other strata encountered, as well as from active or incipient mine fires.

Dusts found in metal-mine air are derived largely from drilling, blasting and shoveling; from mucking, tramming or dumping rock or ore, or from timbering and hauling. Probably well over 50 per cent of all metal mines have siliceous material in ore or containing walls and hence have siliceous dusts, which, as far as is known, are the most dangerous of all of the ordinary earth dusts, especially when taken into the lungs in large quantities in the finely divided form thrown into the air by drilling, blasting and mucking. Certain siliceous dusts seem to have far less injurious effects than others of essentially the same composition. Dusts of other siliceous character, while not considered to be so definitely and immediately dangerous, are nevertheless likely to be harmful ultimately, especially if present in the air in very finely divided particles and in large quantities. The dusts of certain soluble lead ores affect workers through skin absorption as well as through breathing. Dust in coal mines is caused by processes somewhat similar to those in metal mining, in addition to those created by undercutting, and shearing, and much dust is now being created by various kinds of mechanical loading devices. Coal dust may be very explosive under some conditions, and in addition is a definite health hazard especially when present in the air in large quantities during a considerable part of the time spent underground. Moreover, its presence in large quantities in air very greatly reduces visibility and this, in turn, adds greatly to the danger of accidents.

Mine ventilation was discussed much more 15 to 20 years ago than it is now, yet today there are tendencies in connection with mine ventilation which indicate that the lessons learned by bitter experience in the past either have been forgotten or have not been brought to the attention of the present generation. Possibly the mining men of today consider that conditions have changed so much that experiences of the past cannot be repeated. At any rate, it seems desirable to restate some of the principles of mine ventilation that were understood more or less in the past and to call attention to some dangers in connection with present-day trends or practices.

One of the more seriously dangerous trends in mine ventilation is the placing of the main fan underground in coal mines—a decidedly dangerous practice and one very likely to cause numerous disasters if it becomes widespread. The placing of the main fan underground in metal mines is somewhat hazardous but is by no means so dangerous as in coal mines. In many recent main-fan installations on the surface of coal mines the fan has been placed in a direct line with the mine opening—a hazard, since the fan is likely to be destroyed or be damaged seriously in case of an explosion. The main fan for a coal mine should certainly be placed on the surface in a fireproof hous-

ing and should be offset at least 25 ft from the line of the mine opening, while explosion doors that can be opened easily should be provided to relieve pressure in case of an explosion. In many metal mines (partly to reduce air dustiness and partly to prevent formation of ice in working shafts) ventilation systems are installed with the working shaft on return air. This is extremely hazardous where the workers must be transported through the shaft, because in case of fire the shaft is very likely to be filled with poisonous gases and the men in the mine may be trapped or killed. Safety demands that the man-hoisting shaft be safeguarded by being in either intake or downcast air with provision for preheating it in winter. Preferably, the man-hoisting shaft should be essentially neutral as to air flow, with only enough fresh air to keep it free of gases or undue cold or humidity; and other shafts or openings should provide fresh air and remove vitiated air.

Most mine-ventilation installations are intended to achieve only some of the foregoing objectives and consequently fail to give protection to mine workers or mines. In numerous cases a slipshod or poorly planned and operated ventilating system is a menace to either safety or health, or both. However, many of the worst disasters in both coal and metal mines have been due to glaring defects in elaborate and relatively costly mine-ventilation systems and installations or in their careless or ill-considered operation.

GENERAL CONCLUSIONS

Some general conclusions are enumerated as follows:

1. In mining, it appears that ventilation, fire protection and prevention, health, safety, and efficiency are very closely interrelated and, consequently, conditions, installations, and equipment affecting any one of them may have a vital effect on one or all of the others, detrimentally or otherwise.

2. There is at least as much reason for providing adequate ventilation for most metal mines as for providing ventilation for coal mines, though metal mine operators rarely provide ventilation until forced to do so by some untoward conditions or occurrence. Coal mines, on the other hand, are usually ventilated to some extent from the beginning. The increased attention to occupational disease, especially when caused by dust, has provided an incentive for greatly added attention to ventilation by the larger metal-mining companies in the past few years.

3. Every coal or metal mine should have a mechanically driven fan or fans preferably placed on the surface in a fireproof housing and capable of reversing the direction of flow of air currents with a minimum of delay. Metal as well as coal mines should provide fan ventilation from the start of opening to avoid dangers from explosives or other fumes, dusts, etc., and to provide fresh air to workers. Mines that rely on natural ventilation are likely to have periods when air circulation is sluggish, ceases utterly, or reverses in direction; and in case of fire so-called naturally ventilated mines are almost invariably at a decided disadvantage because of inability to control the direction of air flow.

4. Ventilation should be under definite, constant supervision, preferably of one person, who should report to the highest officials. Each mine should be ventilated wholly within itself as interventilation of mines is likely to be dangerous, inefficient and generally unsatisfactory, even though it may have some minor or transitory advantages. Any mine employing more than 200 persons should have at least one person working exclusively on ventilation if health, safety and efficiency are to be protected adequately.

5. Ventilation is probably the most effective means of obtaining dust-control, although it ranks second to wet methods under some conditions. Where ventilation can be applied effectively,, dust can be removed from the air, or its concentration can

be kept so diluted as to render it almost harmless to workers. Ventilation can minimize dust hazards of almost all kinds, including spontaneous combustion, but it may enhance the hazards of dust explosibility in some instances. Ordinarily ventilation should be induced by exhaust methods, but sometimes forced or blowing systems are the more effective, particularly in underground work in places having high-temperature air conditions. It is usually difficult to attain really effective ventilation in such work as tunneling and metal mining because ordinarily openings are driven singly instead of in pairs, as in coal mines, where fresh air flows in one opening and return or contaminated air flows out through the other. Conditions caused by dust may force the use of parallel openings in many future tunneling and metal-mining operations. Notwithstanding the violent objection this suggestion will raise, there are good reasons for believing that in many cases there would be definite monetary savings if the parallel-opening system—with its numerous ventilation, health, drainage and other advantages—were used instead of the present one-opening system, which has numerous disadvantages and hazards and only the one advantage that its use has been customary.

6. Dry drilling is unquestionably the dustiest work in mining, tunneling, or quarrying, and it continues to be used far too much in the United States not only in small mines but in some of the larger ones working in rock or ore having so-called harmless dust. If dry drilling is allowed at all, it should be done only with auger twist drills or with the simultaneous use of dust traps where the use of such equipment is feasible. Wet drilling, as practiced in the United States with water forced or drawn through the drill steel by means of compressed air, for several years has been under fire as producing relatively large quantities of dust, even though the material floating in the air usually is wet. Some of this criticism is undeserved, but some is legitimate. The trouble is due largely to the fact that the holes are collared or started dry and, in some cases at least, are drilled dry for several inches after starting, and the dust created by this dry starting impregnates the usually stagnant air of the ordinary underground-tunnel or metal-mine working place for much, if not most, of the working shift. In some cases the driller limits drastically the amount of water sent through the drill steel and, while drilling is nominally wet, the actual practice closely approximates dry drilling with compressed air flowing through the drill steel and forcing out of the drill hole and into the air of the working place undue quantities of water-entrained dust particles. In some cases, when the drill is out of repair, the drilling allows far too much compressed air to flow through the drill steel. When present types of wet drills are used properly, the dust from drilling can be kept within reasonable limits. This practice is actually being followed by numerous progressive and conscientious mining and tunneling companies at the present time. Some companies in the United States and Canada are experimenting with rock drills that do not allow compressed air to go through the drill steel but require the water to be forced through the steel by its own pressure. Such drills are said to produce less dust than the usual types in which water is sent through the steel by compressed air; but the newer type of drill has been little used in the United States. However, air dustiness can be kept well within safe limits if (*a*) modern types of wet drills are operated with care and judgment, plenty of clean, dirt-free water is used in collaring and the drills are kept in good repair; (*b*) the proportions of water and compressed air flowing through the drill steel are regulated adequately; (*c*) the exhaust from the drill is prevented from impinging on nearby dusty surfaces of muck piles, floors, walls, or timbers, thereby throwing dust into the air; and (*d*) insofar as feasible, the drilling of upper holes is reduced to a minimum, as upper holes, whether drilled dry or wet, are relatively heavy producers of dust in the air of working places.

7. Although there are regulations in most of the states to prevent dust formation in drilling, these regulations are not always observed, especially in small mines or prospects. Although miners recognize the dangers from dust, they often prefer to take the risk rather than endure the slight discomfort or extra trouble of using precautionary methods or devices essential in wet drilling procedure. Mine and state officials seem to feel that unless the miners willingly aid in protecting themselves, the management should not be required to force them to protect their own health and, incidentally, that of their families. Dust-prevention devices of proved success, such as modern self-rotating wet stopers, should entirely supplant dry drills, and their use should be enforced upon both miners and operators in metal mines, large and small, irrespective

of the silica content of the rock being drilled. There is absolutely no valid excuse for dry drilling in modern metal mining, and to date no workable device is available for removing dust formed in dry drilling in underground mines.

8. Next to dry drilling, blasting causes the greatest quantity of finely divided dust to be thrown into working places through bringing down rock in mining, quarrying, and tunneling. In addition to throwing violently into the surrounding air immense quantities of finely divided dust, blasting, with the heavy charges of explosive required in many mining and tunneling operations, also impregnates the surrounding air as well as the muck piles with considerable quantities of poisonous gases, such as carbon monoxide, oxides of nitrogen and, in some kinds of rock, with hydrogen sulphide and other dangerous sulphurous fumes. Practically all of these occur in such quantities and percentages under some circumstances as to asphyxiate or cause serious illness in persons who may breathe them. Many persons who have studied the incidence of dust disease believe that the breathing of even small quantities or percentages of extraneous or harmful gases, such as carbon monoxide, oxides of nitrogen, hydrogen sulphide, etc., inflames or otherwise adversely affects the respiratory organs, especially the lungs, and makes them much more easily and readily susceptible to harm from the breathing of dust particles.

9. Blasting practice must be modified and reformed if employers are to avoid heavy penalties in compensation and other charges due to legislation on occupational diseases (one being dust disease) in the various states. If at all feasible, blasting should be done at the end of the working shift or on an off shift, and the dust- and gas-laden air should be removed or thoroughly diluted before men return to work. Ventilation is an essential agency in cleansing the air where blasting has been done; but, in addition to being ventilated, the place should be thoroughly wetted (walls, floor, roof or top, timbers, etc.) before blasting is done, and in metal mines a water blast should be used during and after the blasting. The region should be wetted thoroughly upon return to the face after blasting, and the muck pile should be kept well wetted at all times while it is loaded out, as the water not only lays the dust but also either absorbs or otherwise aids in diluting or eliminating harmful or poisonous gases which follow blasting and which usually cling to the blasted material. In general, the larger the quantity of explosives used and the more finely divided the blasted material happens to be, the more heavily will the air be impregnated with dust and smoke and the greater will be the quantity of poisonous gases left at the place. Methods used should therefore be such as will tend to reduce the quantity of explosives used and the gases and dust produced. In this connection, it will be found that the use of stemming or of some types of blasting plugs now available will confine the explosives more definitely, will increase their efficiency and reduce the quantity of explosives necessary, will reduce the volume of poisonous gas produced, and possibly may decrease the violence of the blast and reduce its tendency to raise and disseminate dust. It would be well for users of explosives to investigate types of explosives known to emit minimum quantities of harmful gases on detonation and to use them. Discontinuance of the use of fuse and substitution of electric blasting would aid in reducing the amount of poisonous gases caused by blasting. Explosives that have been held in storage too long or have been stored under unfavorable conditions as to moisture, temperature, etc., are likely to give off maximum quantities of harmful gases. The utmost care should, therefore, be taken in the storing of explosives to see that they are stored under favorable temperature and humidity conditions and are not held too long before being used.

10. Although finely divided dust in mines is probably the chief cause of miners' consumption, it is now recognized that other factors, such as high temperatures and humidities, harmful gases, and lack of air movement, may have almost equal influence. All of these factors can be controlled readily by ventilation. It appears that with a dry-bulb temperature below 75 F, mine working places may be comparatively comfortable, irrespective of the air movement or relative humidity. However, the presence of air depleted of oxygen (say below 18 per cent) or impregnated with even small proportions of gases, such as CO_2, CO, or oxides of nitrogen, any or all of which may be produced in blasting, may produce uncomfortable or unsafe conditions.

11. With a dry-bulb air temperature above 75 F, comfort and maximum working efficiency can be obtained only when the air is moving, especially if the air has high relative humidity. The exact velocity necessary is a variable, depending largely upon

the temperature and humidity. Saturated atmospheres relatively free of harmful gases or dusts, up to nearly blood temperature, may be made endurable and even tolerably comfortable by providing sufficient velocity or movement.

12. In still air in metal mines with a temperature of about 85 F and 90 to 100 per cent relative humidity there is likely to be little or no harmful effect on persons completely at rest; but, with even moderate work, body temperature is likely to rise above 100 F, blood pressure may fall perceptibly, and pulse beat may rise materially. In still air, with temperatures of 90 to 100 F and with more than 90 per cent relative humidity, even when the individual is practically at rest, body temperature rises quickly (in cases reaching over 102 F), blood pressure is likely to fall rapidly, pulse beat may increase abnormally and be very sensitive to even slight exercise, and perspiration may be very profuse. Dizziness, physical weakness, mental sluggishness, and headaches are likely to be experienced in time. When even light work is attempted, these symptoms are likely to be augmented greatly.

13. Relative humidity, even up to the saturation point, does not appear to be harmful to health, comfort, or efficiency until the temperature runs above 75 F, and if sufficient air movement is supplied, high relative humidity is not particularly harmful until the temperature is well over 90 F.

14. With the exception of blind-end unventilated working faces, mine air in general is not particularly deficient in quality. However, the air at blind-end unventilated faces of drifts, crosscuts, raises, winzes, and stopes in metal mines is likely to be deficient in oxygen and high in nitrogen or CO_2 and possibly in CO, oxides of nitrogens, or other impurities. There are on record many cases of asphyxiation from these gases in metal mines. Similarly, dead-end unventilated places to the dip, to the rise, or on the level in coal mines may become depleted of oxygen through absorption by the coal or timbers or through intrusion of extraneous gases, nitrogen, carbon dioxide, methane, etc. Places under seal in either coal or non-coal mines are almost sure, in time, to have the oxygen content considerably depleted and the carbon dioxide or nitrogen (or both) content increased.

15. In stagnant air comparatively small quantities of impurities, such as 0.30 per cent or more of CO_2, 0.02 per cent or more of CO, or oxygen slightly below 20 per cent, cause headache, dullness, etc., and this is particularly true when the temperature is above 80 F. However, these small quantities of impurities are not so likely to be noticeable when there is perceptible movement of the air.

16. Frequently blind-end actively worked faces in mines (and more particularly in metal mines) have air so depleted of oxygen that a candle will not burn and carbide or other lamps must be used; hence, oxygen content is below 16 per cent and CO_2 may reach several per cent. Occasionally entire mines are found with this condition, which is regarded as satisfactory by many mine managers. Men working in an atmosphere that will not support the combustion of a candle cannot work with maximum efficiency, and ultimately their health will probably be affected. Safety is certainly endangered by such conditions, inasmuch as human life cannot be sustained when the oxygen content of the air is less than 6 per cent.

17. Mines with cool working places that allow men to work at top speed, especially when contracting, are likely to be extremely dangerous to the health of workers unless provision is made to remove explosives fumes and other gases and fine dusts from working places with ventilating currents. Mines with high temperature (above 75 F) and high humidity (above 85 per cent) are likely to reduce the efficiency of workers 25 to as much as 75 per cent, and workers are likely to become unhealthy eventually unless moving currents of air are supplied to working places. Unhealthfulness and inefficiency are hastened and intensified if fine dust, especially siliceous dust, is present and if blasting is done, especially when men are in the mine.

18. Many accidents in mines are due to deficient ventilation. Failure to remove smoke and fumes from explosives prevents efficient inspection of working places to make them safe. In addition, many men have been asphyxiated by fumes from explosives. In hot, humid, stagnant air men are likely to be affected by dizziness or by lack of ability to think clearly or quickly, or they may faint and be killed. Also, there are numerous instances where men have been known to have dropped dead in hot places, and scores of underground workers have been killed in unventilated

places where the oxygen content of the air has been reduced in any of several ways previously mentioned when ventilation has not been maintained.

19. When the temperature of the air exceeds 75 F. mere movement of the air at working places is more important than anything else in giving adequate ventilation, provided such air is reasonably free of noxious gases and dusts.

20. Flowing air in underground passages rapidly assumes the temperature of the surrounding rock. The rate of change is variable and frequently is as high as or higher than 1 deg F for every 100 ft of air travel. The temperature of still air underground rarely varies more than a few degrees from that of the surrounding rock or water. Rock temperature generally increases with depth, the rate of increase ranging from 1 deg F or more per 100 ft of depth in certain districts of the western part of the United States to but ½ or ⅓ deg F per 100 ft of depth in other regions, both of the United States and of foreign countries. Rock temperature may vary at the same depth in different kinds of material. A copper sulphide ore with quartz gangue in a mine in the western part of the United States had rock temperature several degrees higher than a zinc sulphide ore in quartz gangue in a parallel vein about 200 ft distant, both on the same level and both practically free of water.

21. Water standing still or flowing on the floor in mines readily communicates its temperature to surrounding air. Water dripping through the air quickly brings the air practically to the temperature of the water drippers, and profuse water drippers will determine the temperature of the air almost irrespective of rock temperature. Water temperature underground is generally, but not always, the same as that of surrounding rock.

22. The dry-bulb temperature of air passing through fans frequently has increased several degrees Fahrenheit and the relative humidity has automatically decreased. Small, electrically driven fans with galvanized-iron, canvas or other flexible tubing are being used widely to great advantage in metal mines and tunnels to carry air to dead ends. The galvanized iron has the advantage of allowing reversal of air currents to pull smoke out after blasting, then to force moving air to workers after the smoke has been removed. Moreover, it does not deteriorate as fast as canvas. In general, the flexible tubing must be used only in forcing air to the face. Its advantages are low first cost, ready installation and removal, flexibility in conforming to bends or turns, and ease of repair. Moreover, because of its ready installation and removal, the flexible tubing can be brought close to the working face at ordinary times and easily removed before blasting to prevent its destruction. Either method readily permits 500 to 5,000 cu ft or more of moving air per minute to be applied at the working face at comparatively small cost. The use of these small fan-tubing units in underground coal mines is hazardous in the extreme; and in the past two decades scores of explosions, with heavy loss of life and many fires, have been caused in coal mines through the use of fan-tubing blower systems.

23. Compressed air from the end of air hose is used extensively to remove explosives fumes from faces or to ventilate hot, stagnant, blind-end workings in metal mines. These blowers deliver about 100 cu ft of air per minute, but its temperature rarely varies much more than 2 or 3 deg from the temperature of the rock and air of the working place. Such compressed-air blowers are inefficient for removal of smoke or gases, provide comparatively little pure air and cause very little reduction in the temperature of the surrounding air. Moreover, it costs about 100 times as much to place 1,000 cu ft of compressed air at a working place as to circulate a like amount of air by ordinary ventilation methods. Compressed air is employed only to a limited extent in ventilation in coal mines and its use in coal mines is not only expensive and inefficient, but also definitely dangerous.

24. Air in mines is cooled by using ice or sprays of cool water, refrigeration, rapid coursing of air brought from the surface or carried slowly through workings with cool walls, excluding air from hot, abandoned workings and excluding return air from currents of active workings and possibly other procedures. Water sprays are not nearly as much employed as they should be. Ice is used to a slight extent in the United States. Refrigeration is costly and found in a very few mines, although several hot metal mines in the western part of the United States are now using refrigeration to advantage. Rapid coursing of air currents from the surface, which can be brought about most efficiently by the establishing of definite splitting systems, is used only occasionally, though it is quick, relatively cheap, and efficient. Failure to

seal abandoned places having decayed timber and hot rock or water sends much unnecessary heated and vitiated air into the mine, and re-using the return air has the same effect. In some mines the cold, fresh air from the surface is coursed along drainage tunnels taking relatively hot water out of the mine in open ditches, thereby transmitting higher temperatures to the ingoing fresh air.

25. At time of fire in a mine, lack of an efficient ventilation system may be disastrous. Each mine should have at least one main fan, which should be placed so as to be inaccessible to fire; have fireproof housing; be capable of quickly reversing the direction of flow of air currents if desired; and, preferably, be installed on the surface. There should be a definite system of air splits, such that fire in one place may not necessarily fill the entire mine with poisonous fumes. This provision is vitally important, yet very few metal mines have made even a reasonable attempt to establish this excellent safety feature. There should also be a system of doors near shafts in the levels leading from shafts, such that the entire shaft may be isolated readily in case of fire, or any part of the mine may be isolated from the shaft. If the main haulage road or hoisting shaft or slope is used to transport workers into or out of the mine, it should not be on return air; otherwise, lives are very likely to be lost because of poisonous gases in case of fire or explosion. The ideal system as to mine openings through which men travel or must be hauled or hoisted is to have such openings essentially neutral as to air flow and to supply them with only enough intake or fresh air to keep them free of gases, unduly cold air, etc.

26. Experimental work in mines reveals that after shafts that previously had ordinary exposed timbers are smooth-lined, friction generally will be reduced to such an extent that 50 per cent or more additional air can be handled by the same power. Preferably, at least the intake air shaft should be fireproof. If possible, all shafts or heavily inclined openings that carry air or through which men travel or are transported should be fireproofed, at least at the intersection and within several hundred feet of the intersection of such mine openings with the surface.

27. Although the cost of establishing a ventilation system for a large mine is variable, the expense of operation is not particularly burdensome and usually will be offset in a metal mine by savings in compressed air and by an increase in efficiency and health and safety of employees, which frequently will equal the entire cost of the investment within a few years. If a fire or explosion occurs, an efficiently installed and operated mechanical ventilation system is of incalculable value, and the absence of such a safeguard is likely to result in heavy loss of property and possibly of life.

CONCLUDING REMARKS

Ventilation has always been a primary requirement in the operation of underground coal mines, chiefly because of the possible occurrence of explosive gas, and virtually all of the countries of the world as well as the several states of the United States include ventilation requirements in the laws concerning coal mines. Ventilation requirements, legal or otherwise, for non-coal mines in the United States are and always have been inadequate. The state laws and state regulations as to the ventilation of coal mines fail to cover the ventilation requirements for health, safety or efficiency. Changing conditions in the relatively recent past have caused metal mines to do a much more effective job of distributing suitable quantities of relatively pure air to their underground employees, while on the other hand, recent changes in coal mining have more or less automatically decreased the ventilation efficiency of the coal mines of the United States rather than increasing it or even keeping pace with the trends of the times.

Many metal mines have acknowledged the trend toward the enactment of increasingly drastic laws concerning occupational disease (more especially dust disease) and have recognized that ventilation is one of the most efficient of the available methods of minimizing its occurrence. A number of large under-

ground metal mines are now doing a fairly effective job of supplying underground workers with air of a suitable quality as well as in sufficient quantity to serve the needs of health, safety and efficiency.

The trend toward substitution of mechanized mining of coal for much slower *hand* methods of the past has had definitely detrimental influence on the effectiveness of ventilating coal-mine working faces; the result is apparent in the fact that practically all of the coal-mine disasters (fires or explosions) for the past several years have occurred in highly mechanized mines. The cause is readily apparent, but the remedy is not yet at hand. Mechanized mines cause the working faces to advance several times as rapidly as with the hand loading methods. This has very greatly increased the liberation of explosive gas and of finely divided (hence, highly explosive) coal dust. At the same time, makeshift ventilation methods have been introduced, such as the use of small portable blowers and flexible tubing. The result has been anything but satisfactory. The workers' lives are being sacrificed in far too numerous mine disasters because excessive volumes of explosive gas have been liberated and provisions for carrying it away are ineffective. Moreover, sources of ignition of gas-air accumulations have increased. Also, the dust content of the air in many of the mechanized working places under the greatly accelerated extraction of coal has become a menace to the health of those breathing it as well as a menace to their safety due to marked lessening of visibility in dust-laden air.

Another serious problem connected with coal-mine ventilation is the fact that with mechanized mining procedures, mines open vast areas of coal in a relatively short time, soon placing the working faces long distances (in some cases as much as several miles) from the mine openings. In many mines coursing of air in sufficient volumes through these long distances is difficult and in others almost impossible; hence, there is now a crying need for additional openings to the surface relatively close to the mine working faces. Newer methods of sinking circular 36- to 72-in. diameter shafts at relatively low cost may solve this particular problem, which certainly becomes more acute daily.

Mines must be adequately ventilated. Present conditions prove that more effective ventilation of our mines—especially those that produce coal—challenges the ingenuity of mining people and constitutes a worthwhile post-war project. Unfortunately, relatively little can be done during the present great demand for production.

DISCUSSION

A. C. BARTLETT, Boston, Mass. (WRITTEN): The opening statement in this paper wherein the author states that *Ventilation is the keystone of safety in underground coal mining, and of health in underground metal mining,* cannot be too strongly stressed, and the frequency with which headlines appear in the daily papers telling of mine disasters due to mine explosions could, in the writer's opinion, be greatly reduced by adequate ventilation.

The author very aptly compares a mine to a large building, and if we compare it to a windowless building, I believe that all of our members who are consulting engineers will agree that adequate ventilation cannot be obtained by merely creating an air movement along the corridors. For instance, how long would the occupants of the Pentagon Building in Washington continue to carry on their work if all outside

windows were blocked, and the only ventilation provided was that due to movement of air along the corridor at a velocity of 1000, 2000, or even 5000 fpm?

In some mines, small auxiliary blowers are used to take air from the main corridors and force it through flexible tubing to the working faces, but these are almost universally of the same size, having a capacity of from 1500 to 2000 cfm, regardless of the size of the drift, and regardless of the number of men working at the face.

In my opinion, mine owners and mining engineers concerned with mine ventilation might well take a lesson or two from industrial engineers, who must solve ventilation problems encountered in the manufacture or use of explosives and explosive gases.

Many states, in which mining is a major industry, have adequate laws or regulations relative to mine ventilation but seem to have inadequate facilities for enforcing those regulations. Often the regulations are too general in their scope, and inspectors too lenient to compel the operators to carry out even the intent of the regulations.

The author points out that 15 or 20 years ago this subject of mine ventilation was discussed more than it is today. I believe this to be a fact, although, in my opinion, ventilation is more important today than it was then, and should be discussed even more.

It is my opinion that the lack of discussion is a result of unscrupulous competition which forces the operator to purchase inadequate ventilation equipment if he is to stay in business at all. This brings to mind that in those days of 15 and 20 or more years ago, and even today, it was not uncommon to see some ventilation equipment installed which, if tested by the Society's standard, have to show mechanical efficiencies in the range of from 93 to 100 per cent to equal the performance expected of it by the operator.

We also, perhaps, do not hear so much about respiratory diseases in mines as we formerly did, but that is probably no indication that they are less prevalent at this time than they were years ago, especially in mines where dry drilling is done. Although this part of the problem, in my opinion, is not so closely connected with the problem of ventilation as might be expected at first consideration, it is nevertheless a problem to which the mine operator should give careful consideration.

In localities where extensive cutting of granite and other stone is done, it is not uncommon to find regulations which require the installation of dust removal and collecting equipment, even though the operation is conducted out of doors. The problem is considerably worse in metal mines where the drilling is done in comparatively close quarters.

With the modern equipment designed for collecting dust, particularly in small units, it would seem that it would not be too much of an imposition on the operators to provide the drillers with a unit dust collecting system which would, in all probability, remove at least 80 to 85 per cent and perhaps 90 to 95 per cent of the siliceous dust from the air.

The use of compressed air for ventilation in blind ends or where drilling is done, is extremely inefficient as the author of the paper points out, and at the best, can only serve to stir up the dust and bring it to the breathing level of the miner.

Air conditioning in mines is not uncommon, particularly in the gold mines of South Africa and in some of the other rare metal mines, and is almost a necessity for deep mining operations. The whole field of mining offers a challenge to ventilating engineers to bring about working conditions in mining industries comparable to those enjoyed by workers in other industries because mining, at the present time, is not only hazardous from the viewpoint of explosion and bodily health hazard, but it is also, in a great many cases, an extremely uncomfortable type of labor for the miner who has to carry on the actual work.

Again referring to the comparison of a mine with a building, it is not such a difficult matter to convince an architect who is designing a building, that he must provide certain areas and certain shafts for the location of ventilation apparatus, ducts, etc., but it is sometimes a considerable undertaking for a mine owner to provide a sepa-

rate ventilating shaft for the operation of ventilating equipment when the shaft has to be sunk anywhere from 2000 to 10,000 ft for that purpose.

The practice of ventilating a mine through the same shafts, corridors and drifts that are used for the passing of the material as well as for the transporting of the miners should be discouraged, particularly in mines where the danger of explosions is present. Unless the direction of air flow in such an installation can be changed almost immediately in case of an explosion, an ensuing fire, instead of being confined to a certain area of the mine, may be spread over a very much greater area by the air currents.

It is my opinion that a great deal could be accomplished by additional regulations in some states, and by judicial, yet forceful application of already existing laws and regulations. Many mine owners, fortunately, do not object to the installation of proper ventilating material, if they are properly advised as to what the equipment really should do, and if they are given honest values and honest information.

AUTHOR'S CLOSURE: Evidently Mr. Bartlett read my paper carefully and I appreciate his interesting comments. In addition to the rather elaborate systems in some of the gold mines of the Rand in South Africa, a few of our copper mines have done some excellent air conditioning work; notably the Anaconda Copper Mining Company in some of its deep mines in Butte, Montana, and the Magma Copper Company at Magma, Arizona.

The reference to the greater use of small individual dust collecting units for underground mine working places or faces, especially in regard to drilling, calls attention to the fact that several attempts have been made to use such systems, but up to date they have not proven successful. The problem of air dustiness from drilling is being met fairly successfully by the more widespread use of wet methods, though too much dry drilling is still done. A more recent (and probably in time more effective) procedure to reduce mine air dustiness due to drilling is the use of diamond drilling for blasting holes rather than using compressed air percussion drilling. Diamond drilling of blasting holes is in its infancy but it appears likely that its use may be greatly extended and, if this is done, air dustiness due to drilling of blasting holes will be negligible.

Respiratory disease is rapidly being eradicated from metal mines because the former ostrich-like attitude of metal mining people has been thrown aside. At present nearly all large metal miners are taking due precautions, something that very few were doing 20 or more years ago. Respiratory disease in coal mines is beginning to be recognized as a real menace and some of the more wide-awake coal mining companies are taking cognizance of the hazard. They are trying to find the causes and to apply remedial action.

1277

No. 1277

TOBACCO SMOKE CONTROL—A PRELIMINARY STUDY

By Charles S. Leopold,* Philadelphia

HIGH TOBACCO smoke concentration in auditoriums in general, and in a sports arena in particular, results in interference with visibility, eye irritation, and certain odor effects.

The *interference with visibility* may be a direct obscuration of object viewed due to opacity of the intervening space, or a distraction and interference with vision due to the illuminated smoke cloud.

Eye irritation effects are: smarting and lachrymation.

Odor effects may be divided into: perception of odors upon entering the space, residual odors in clothing after leaving the space, and odors so pronounced as to be objectionable during occupancy.

Interference with vision is a function of the concentration of particulate matter, the distance from object viewed, and the physical relation of the lighting system, stage and seats.

Eye irritation depends upon the irritant concentration in the smoke and, with a simple ventilating system, is a function of the smoke density. Both fluid and particulate matter may contribute to eye irritation or the particulate matter may act solely as a conveyor of irritant gases or liquids. The smoke components may, and probably do, differ qualitatively and quantitatively in their effect on eye irritation and odor.

An electrostatic precipitator can remove a large portion of the particulate matter from smoke. As yet, we do not know how much of the eye irritant and odor producing substance is simultaneously removed.

Odor producing substances are similarly proportional to the smoke density. Charcoal adsorbers are effective in the removal of odor causing substance and have little effect in reducing fine particulate matter. An air washer with clean water will remove some odor but has little effect on fine particulate matter.

Dilution of the tobacco smoke by smoke-free outdoor air is effective in reducing all of the discomforts listed, but in year-round air conditioning systems, the amount of outdoor air required may prove economically excessive.

Where apparatus for smoke reduction is introduced in the recirculated air stream, the available apparatus, in general, does not deal equally with the three types of smoke difficulty. In speaking of smoke-free air as applied to a conditioning or ventilating system which utilizes smoke reducing apparatus in the recirculated air stream, it is necessary to differentiate between *particle-free, irritant-free,* and *odor-free air.*

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* Consulting Engineer. Member of A.S.H.V.E.

Presented at the 51st Annual Meeting of the American Society of Heating and Ventilating Engineers, Boston, Mass., January, 1945.

QUALITATIVE OBSERVATION AT MADISON SQUARE GARDEN

For the seating and lighting arrangement used in the Garden, for basketball or boxing, and with fans delivering a limited quantity of outdoor air at a constant rate, without treatment other than heating, the smoke problem is presented as follows as the number of spectators increases: A general haze is quickly noticeable. The smoke becomes quite visible in the strong beam of the spotlights. Vision from the observer to the spectators at the far end of the arena, when viewed across the ring, becomes obscured. Awareness of the cloud in the spotlight becomes objectionable. Eye irritation is noted. In the

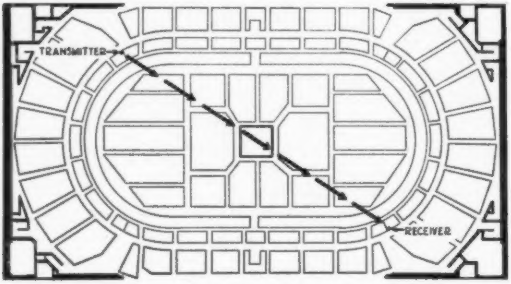

FIG. 1. LOCATION OF OPTICAL EQUIPMENT

practical range of ventilation, odor is not noticeable to the acclimated spectator, but it may be briefly noted by the patron entering the arena.

SELECTION OF TEST METHOD

Analysis of the problem indicated that a measurement of light transmission through the smoke cloud in an auditorium with a simple ventilating system would provide a reproducible scale of smoke density to which observations on visual discomforts, eye irritation, and odor could be referred.

Test Method

A narrow beam of light of constant intensity was projected diagonally across the arena as shown in Fig. 1 and the intensity of the beam at the end opposite to the projector was measured by photoelectric means. The receiving photoelectric cell was shielded so that it would not respond to light other than the projected beam.

Optical transparency is defined [1] as the ratio of the intensity of the transmitted light to that of the incident light. Within the scope of these tests trans-

[1] The Principles of Optics, by Hardy and Perrin (McGraw-Hill); Photography—Principles and Practice, by C. B. Neblette.

parency is used to denote the ratio of the transmitted light with a smoke cloud to the transmitted light without a smoke cloud.

Opacity is defined as the reciprocal of the transparency.

Density is defined as the Log_{10} of opacity, or Log_{10} of $\dfrac{1}{\text{transparency}}$

The relation of transparency, opacity, and density is shown in Fig. 2. The concept is further illustrated in Table 1, in which the relationships are shown for a constant source of light passing through 0, 1, 2, and 3 equal light obstructions, each obstruction permitting the transmission of 10 per cent of the incident light.

TABLE 1

CONSTANT LIGHT SOURCE	A	T	O	D
		1.00	1.0	0
		0.10	10	1
		0.01	100	2
		0.001	1000	3

A = EQUAL LIGHT OBSTRUCTIONS TRANSMISSION 0.10 EACH
T = TRANSPARENCY
O = OPACITY
D = DENSITY

The optical density of a smoke cloud is a function of the number and size of particles that the light beam encounters in passing from the light transmitter to receiver.[2] If it is assumed that the smoke cloud consists of particles of various sizes, uniformly distributed, the optical density becomes a function of the particles per unit volume. In an enclosure with uniform air distribution and uniform emission of smoke, the concentration of particles should be inversely proportional to the quantity of smoke-free air which is introduced. Thus, in comparing various constant rates of ventilation, the optical density would vary inversely as the cfm of smoke-free air per occupant.

The volume of the auditorium and number of air changes *per se* are of interest for an unsteady state, such as the initial build-up of a smoke cloud, or its decline, but not an important consideration in any steady condition, with the possible exception of the combination of extremely low air motion and low air change with light occupancy, in which case adsorption and settlement phenomena may be of practical magnitude.

SIMPLIFICATION OF FIELD TESTS

Since the optical density varies as $\dfrac{1}{\text{cfm per person}}$, it is possible to generalize that for any constant rate of smoke emission the relation of optical density to cfm per person can be plotted on logarithmic coordinates as a straight line parallel to Line 1 on Fig. 3.

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[2] Transmission of Light Through Fog, by F. C. Breckenridge (*Transactions I.E.S.*, XXVII, No. 2, p.216); the Stratten and Houghton formula (Physics Review, 38, 159, 1931).

This generalization simplifies the accumulation of test data as a reading at one sustained rate of ventilation, with a constant rate of smoke emission, is sufficient to establish the actual position of the line and thereby provide a means for predicting the optical densities and transparencies at other steady rates of ventilation.

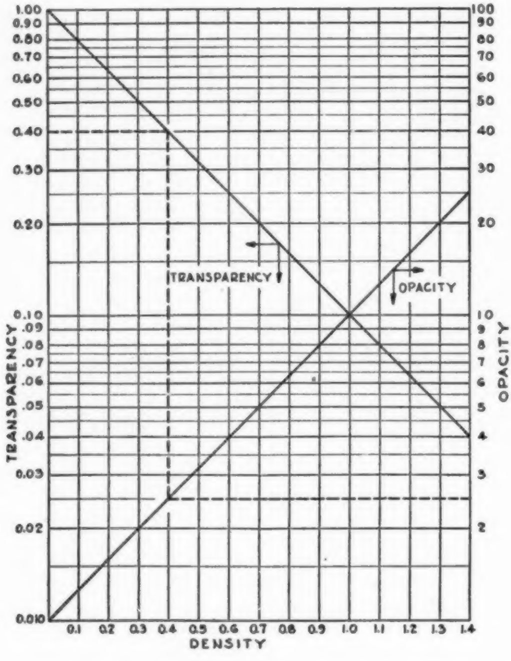

FIG. 2. RELATION OF DENSITY, OPACITY AND TRANSPARENCY

TEST PROCEDURE

A description of the apparatus used is given in the Appendix. The location of the apparatus is shown in Fig. 1. The transmitter and receiver were located approximately 16 ft above the arena floor.

A conventionalized section of the Garden and air distribution is shown on Fig. 4. There are four supply fans on each side of the enclosure. The air discharge of the four systems on one side was obtained from pitot tube measurements. Delivery of the other four symmetrical systems was estimated by a comparison of the static pressure in the supply duct just beyond the fan. Tests were conducted using all outdoor air, untreated except for filtering and

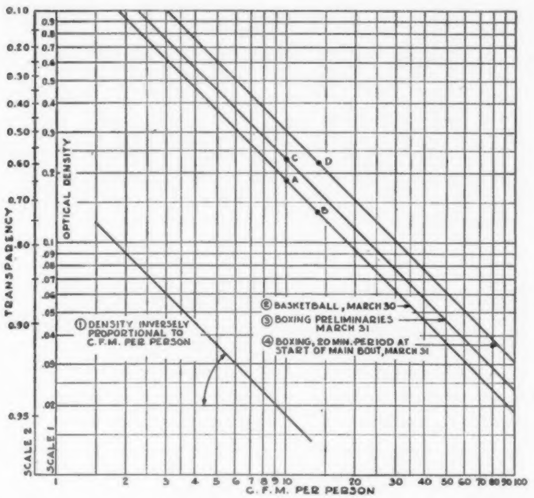

FIG. 3. VARIATION FOR OPTICAL DENSITY AND TRANSPARENCY WITH VENTILATION TRANSMITTER TO RECEIVER 240 FT

heating. The fans, which are provided with 3-speed motors, were run at low and intermediate speeds in these tests. There are eight exhaust fans arranged symmetrically with the eight supply systems, and one additional exhaust fan at each end of the enclosure. All exhausts were operated in conjunction with supply in these tests. There was a slight positive pressure within the enclosure at all times.

Data were obtained for transparency during a basketball game and during a prize fight. In each case readings were started under a condition of light

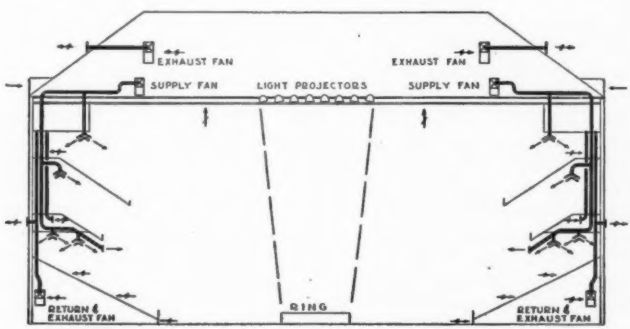

FIG. 4. CROSS-SECTION OF MADISON SQUARE GARDEN INDICATING AIR DISTRIBUTION AND LIGHTING FOR BOXING

occupancy and continued at 10-min intervals for a short time after patrons had left.

Estimates of occupancy were furnished by the permanent personnel of the Garden and were in fair agreement. The maximum occupancy figures are accurate.

Impressions of conditions were noted by five individuals, one a non-smoker and one a heavy smoker. There was satisfactory agreement among observers.

The test data for basketball spectators is shown in Fig. 5. With maximum occupancy, two periods of constant transparency were obtained—one with 10

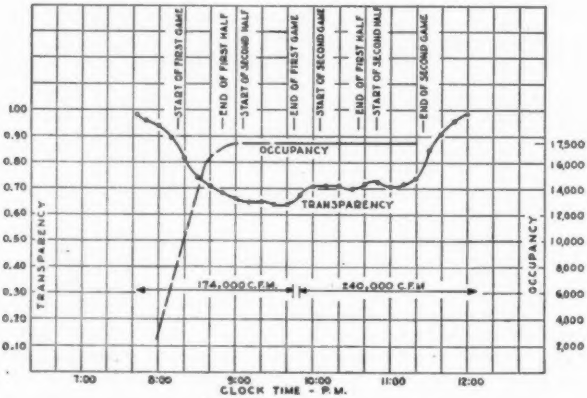

FIG. 5. MEASUREMENT OF OPTICAL TRANSPARENCY (BASKETBALL PROGRAM MARCH 30, 1944)

cfm per person and the other with 13.7 cfm per person. These points are shown as A and B on Line 2, Fig. 3. Within the accuracy of the experiment, the observed data corroborated the predicted relationship of cfm per person and optical density.

Fig. 5 shows the comparison of time, occupancy, and transparency. At 9:50 p.m., fan speed was increased so that the cfm per occupant was changed from 10 to 13.7. Comparatively little time was required to reach the new steady state of transparency. The test indicates that in the practical problem of combating the average maximum smoke density, the air change *per se*, or the lag of the auditorium, is of minor importance. It is of sufficient magnitude, however, to smooth out momentary changes in the rate of smoke emission which, there is reason to believe, occur at times of changing interest in the spectacle.

Fig. 6 is a graphical record of test data for boxing spectators.

Point C, Line 3, Fig. 3, shows the steady state for a peak crowd during the preliminary bouts. A short intermission preceded the main bout, at which

time the ventilation was increased from 10 to 13.7 cfm per person. The increase of transparency, starting during the intermission, was quickly reversed at the start of the main bout and for a period of approximately 20 min the rate of smoke emission exceeded that which occurred during the preliminary bouts. The transparency, however, was somewhat improved due to the increased rate of ventilation. The condition for the steady state during the start of the final bout is shown by Point D, Line 4, Fig. 3.

Figs. 7 and 8 show the data of Fig. 3 on rectangular coordinates.

Figs. 3, 7 and 8 summarize the transparency or density to be expected at Madison Square Garden at a distance of 240 ft, at various conditions of steady

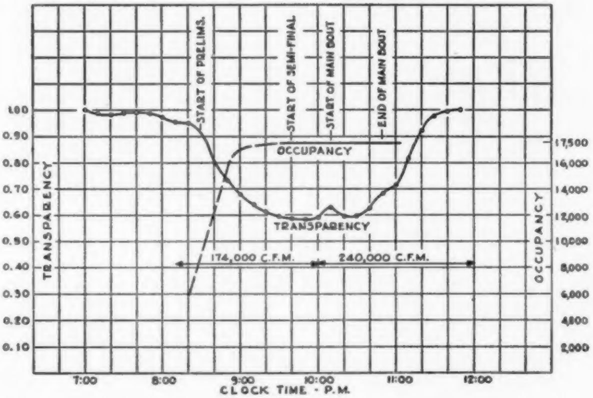

FIG. 6. MEASUREMENT OF OPTICAL TRANSPARENCY (BOXING PROGRAM MARCH 31, 1944)

occupancy and ventilation, when occupied by either fight or basketball spectators. The data pertain to the total number of spectators. Dependable data were not obtained as to the percentage of cigars, cigarettes, or pipes which were lighted, nor for the total number of smokers. A count of only 350 people during a fight indicated that less than 20 per cent had lighted cigars, cigarettes, or pipes. This figure is lower than the values usually assumed and indicates the desirability of further check.

During these tests the consensus of opinion of observers was as follows:

Eye irritation corresponded to a smoke density of 0.167 (transparency 0.675). The visible smoke cloud in the spotlights was judged as *objectionable* at a density of 0.145 (transparency 0.72) as viewed from the arena floor level and 0.670 when viewed from the projector location. Haze, though quite evident, was judged as *commercially acceptable* at a density of 0.097 (transparency 0.8) viewed from the arena. The estimate of *objectionable* is more definite than that of *commercially acceptable*. The qualification *commercially* indicates a compromise. Inspection of Fig. 7 indicates that progressive gains beyond 80 per cent transparency are achieved at rapidly increasing cost in cfm per occupant.

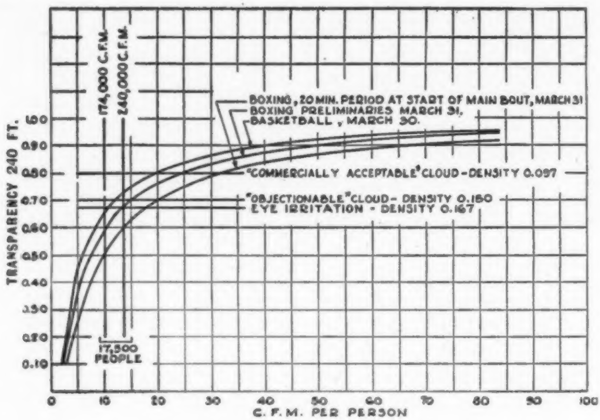

FIG. 7. VARIATION OF TRANSPARENCY WITH VENTILATION

UNIT FOR COMPARISON

The density scale of Fig 3 was calculated from measurements of transparency at 240 ft. Optical density bears a lineal relation to the thickness of the medium and a unit density figure, such as density per foot, can be obtained by dividing the total density by the distance to which the measurement applied. This unit density can then be used for comparison in similar applications.

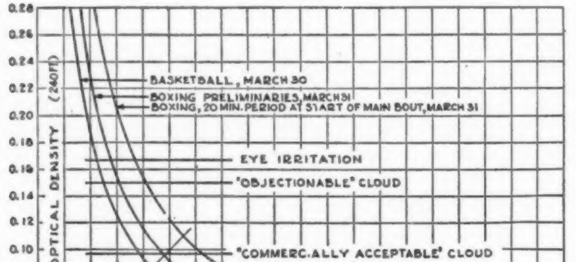

FIG. 8. VARIATION OF OPTICAL DENSITY WITH VENTILATION

Unit transparency would not be so convenient since transparency varies in a logarithmic relationship with distance.

DISCUSSION OF VISUAL EFFECTS

A smoke cloud causes a decrease of light reaching the performer and a decrease in light reflected from the performer to the spectator. The resulting decrease in vision is relatively small at any rate of ventilation which would be adequate for the elimination of other forms of discomfort. Vision, similarly to other human senses, approximately follows the Weber Law[3] to the effect that the change in stimulus required to produce a noticeable change in perception is proportional to the pre-existing stimulus. This is frequently stated in the Fechner[4] form that perception is equal to a constant times the logarithm of the stimulus.

There is a very low level of illumination in the spectator space and any factor causing a decrease in vision would therefore be more noticeable when trying to discern detail across the auditorium than it would be in looking at the brightly illuminated center of interest. There is a further decrease in visual perception due to the light which is scattered from the spotlight beams competing with the stimulus from the center of interest. For very heavy concentration of smoke, decrease of light due to opacity may be of importance.

For reasonable ventilating rates, the most objectionable visual effect is the awareness of the smoke cloud. This visual effect may be partly of psychological origin but it is none the less real. The psychological factors involved are beyond the scope of this paper. The objectionable qualities of this effect depend on the angle formed by the line of sight and the line of the spotlights, and further on the depth of the smoke cloud through and beyond the center of interest. Thus, a spectator looking from the spotlight to the center of interest is less conscious of the cloud than is the spectator looking at right angles to the projector beam; and a spectator on the arena floor, looking across the ring the long dimension of the building, is more conscious of the smoke cloud than is a spectator looking across the short dimension of the building.

From the former observation it would follow that in a motion picture theater, where the sight line largely parallels the line of projection, smoke would interfere very little with vision and this appears to be the case. In motion picture theaters, smoke is quite evident in the beam if the observer looks across the beam rather than at the screen.

The smoke cloud in the Garden was appraised from various positions in the auditorium under a constant smoke density and constant illumination. In the order of decreasing visibility of the cloud, the points of observation were:

1. Lengthwise of the arena at ring level.
2. Crosswise of the arena at ring level.
3. Crosswise of the arena at mezzanine level.
4. Any position from the top balcony.

Lighting projectors are uniformly placed above and beyond the outline of the ring and the basketball court. For this illuminating system it appears that if smoke density is held at a level which will not produce an objectionable light beam for patrons looking at right angles from a distance of approxi-

[3] The Science of Seeing, by Luckiesh and Moss.
[4] Handbook of Engineering Fundamentals, by Eshbach.

mately 100 ft, it will not be objectionable for patrons in the high gallery who have a total sight line of approximately 220 ft. When viewing from any point other than the arena level, the spectator looks down at the relatively bright floor of the ring and immediate surroundings. When looking across the ring from the arena level, the background of the fighters is the smoke cloud against the relatively dark house. As the density of the cloud approaches an *objectionable* condition it is not possible to distinguish detail in the stadium seats beyond the arena when looking lengthwise through the house. Polarization [5] of the scattered light was noted by direct observation with a polarizing filter.

DISCUSSION OF EYE IRRITATION

Previous experiments [6] have indicated that eye irritation is a function of irritant concentration. It would, therefore, be expected that the smoke density at which eye irritation was noted in these experiments would be a constant applicable to other work.

DISCUSSION OF ODOR

The odor problem in the Garden may be subdivided into: odor for the acclimated spectator; odor for the spectator entering the house; odor remaining in the clothes.

Due to rapid olfactory fatigue, there is no practical problem of odor for the acclimated spectator. An odor must be indeed objectionable to a person entering the auditorium to be even perceptible to a spectator who has been inside for a short time. Any quantity of outdoor air which will assure satisfactory visual effects will avoid objectionable odor for the acclimated spectator. It could be of importance in an extreme case, such as the use of a very small quantity of outdoor air in combination with some form of apparatus in the recirculated air system which would remove the visually objectionable smoke without the removal of the odor-causing substance.

A patron going to his seat must first pass through a ventilated lobby. As a rule, smoke concentration in a lobby is somewhat less than in an auditorium, though this condition can at times be reversed. If there are two steps of increasing smoke concentration, odor perception is minimized by the relation of change in stimulus to the pre-existing stimulus and rapid olfactory fatigue.

The factors affecting the odor of tobacco remaining in the clothes have not been adequately investigated. The relative importance of smoke density, time, temperature, and water vapor is not known, either for the period of exposure or for the period which follows.

LIMITATIONS OF THE DATA

The analysis assumes a uniform distribution of smoke and smoke-free air. Smoke appeared uniform when the house lights were on and the spotlights

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[5] Fragments of Science, by John Tyndall.
[6] Noxious Gases, by Henderson and Haggard (Reinhold Publishing Corp.).

were off. No actual measurements were made of the uniformity of smoke or the specific air supply directed toward the ring.

Tests were conducted for the illumination normally used for basketball and for boxing. Experiments were not conducted on the effect of variations in light intensity, either of center of interest or of background.

Four of eight fans were accurately tested at two speeds. Total air quantities should not be depended upon for an accuracy of better than plus or minus 10 per cent. Variation in capacity between the two ventilating rates was confirmed by static pressure readings at all fan discharges and should be accurate as a ratio.

The data represent two evenings of measurement preceded by one preliminary afternoon test. Data appear consistent but, as noted, the period of test was limited.

The specific data of these tests, in general, cannot be safely extended to enclosures with different types of occupancy and different physical arrangements. The test method outlined is relatively uncomplicated and convenient for use in the field. Additional data will be useful and it is suggested that any data subsequently collected be forwarded to the A.S.H.V.E. Research Laboratory to help fill the general void on this subject.

ACKNOWLEDGMENTS

The Precipitron Application Department of the Westinghouse Electric Corp. cooperated in these experiments by providing the optical equipment and the assistance of H. W. Wrigley, Laboratory Supervisor, during the conduct of the tests.

APPENDIX

DESCRIPTION OF EQUIPMENT

The projector consisted of a Westinghouse S No. 787917 Portable Traffic Signal Projector with all unnecessary parts, including handles and switches, removed so that only the housing, lamp and socket, back, primary and secondary reflectors remained. A bow and base capable of being rigidly clamped were added for convenience in mounting. A No. 1503 lamp, 50 candle, 6-8 volt, was used as a light source. With this lamp in focus in the reflector, a narrow beam of light was obtained having a spread of approximately 1½ deg and an intensity of approximately 500,000 candlepower.

The receiver consisted of two selenium type self-generating barrier layer cells mounted in the same type housing as the projector, with two baffles with circular openings adjusted in front of the barrier layer cells inside the housing so that the effective field of view of the cells was limited to a circular area centered at the projector.

The barrier layer cells in the receiver were connected in parallel to a high sensitivity microammeter.

Both the projector and receiver were bolted solidly to sloping steel-surfaced banisters in such a position as to be not readily accessible to spectators. The operators occupied the seats alongside. The location was such that as few people as possible were disturbed by direct light from the projector and no light from any fixture in the building could enter the receiver and produce an erroneous reading.

To make the tests, the projector was focused to produce as nearly as possible a perfectly collimated beam of light, the maximum intensity of which was directed against the barrier layer cells. The sensitivity of these cells was then adjusted to

give approximately two-thirds scale deflection of the microammeter when the air in the building was as clean as possible. The lamp in the projector was operated from a constant-voltage transformer in order to maintain constant light output. Readings of the light intensity reaching the receiver were made every 10 minutes throughout the test period by turning on the projector, allowing 30 seconds operation for the light intensity from the projector to become stabilized, and then recording the meter reading. The projector was turned off between readings in order that the spectators might not be disturbed any more than necessary.

DISCUSSION

F. H. MUNKELT, New York, N. Y. (WRITTEN): This excellent approach to a quantitative determination of tobacco haze and its control by ventilation makes an interesting study. That it was applied to an unusual structure does not detract from its value. There is, at least, symmetry and there are within the building no interfering barriers, corners or partitions which often qualify the value of tests in smaller structures when air distribution is a factor.

It is about 10 years since I first tried to analyze the problem of removing tobacco contamination from the air. It is, therefore, indeed refreshing to have this paper appear. The conclusions which I reached at the time were very simple: that the solution of the problem consists of two parts, first, the removal of the smoke particles and second, the removal of a certain vapor or vapors. My views today are substantially the same although they have been modified somewhat in detail.

The author has distinguished between eye irritation and odor. It is assumed that he has not precluded the possibility of both being caused by one substance. Experience with the tobacco problem points to the conclusion that both the major irritant and the major odor present are characteristics of the vapor of pyridine. Other substances with one or both of these characteristics may be present, but certainly in much lower concentration.

Pyridine is a liquid with a most pungent odor which, in a diluted state, resembles the odor from stale cigars. Its toxicity is reported [7] as "local irritation on the mucous membranes and a general narcotic action on the nervous system. Symptoms following exposure are irritation of the respiratory tract and of the eyes, and cough."

Tobacco contains nicotine, which is a very poisonous oily liquid. Nicotine is a by-cycle molecule, one cycle of which is identical with the pyridine ring. Reducing the mechanism of smoking a cigar to its simplest terms, the heat of the lighted end evaporates some of the nicotine, and at the same time dissociates some nicotine to produce pyridine. Presumably a large fraction of both these high boiling vapors are condensed in the lower part of the cigar, causing it to become ranker as it becomes shorter, and leading our physicians, when they find it necessary to regulate individual smoking habits, to warn against smoking cigars more than half way. Pyridine is the more volatile of the two and would be expected to be the predominant vapor in mixture with the smoke.

The irritant level has been determined by the author in terms of smoke density. It is customary to express vapor concentration in terms of parts per million or milligrams per litre. However, in the case of smoke and vapor emanating from the same source expressing the irritant level in terms of smoke density may be entirely satisfactory as the following general observations will indicate. These are not tests which I have made or which I have found reported anywhere. They come out of my studies in the field of sorption of gases and vapors by solids and they certainly indicate the overlapping nature of the smoke and vapor present in cases of this kind. When a substance is burned, gases, vapors and fine particulate matter called

[7] The Analytical Chemistry of Industrial Poisons, Hazards and Solvents, by Morris B. Jacobs. (Interscience Publishers, Inc., 1941.)

smoke are produced. Of course, in some cases, the quantity of one or two of these products could be zero. The smoke particles, largely carbonaceous and produced under conditions of heat, are active adsorbents to a greater or lesser degree. Any adsorbent will retain a certain minimum amount of a given vapor under most conditions except that of increased temperature. Placed in a high concentration of the same vapor it will adsorb an excess of the vapor. That leads to the belief that the smoke particles adsorb, as soon as cooled and while still not far from the source of combustion, an excess quantity of the vapor, having the characteristic odor of that combustion, and then release the excess slowly after the surrounding concentration has been dissipated. It is one way that the lingering odor of hydrogen sulphur dioxide in soft coal smoke can be explained. It is one way that the lingering effect of the odor of pyridine in tobacco smoke can be explained.

If there is any sense at all in the adsorptive nature of smoke it serves to emphasize the necessity of removing both the tobacco smoke and the irritant vapor of pyridine, if satisfactory conditions are to be maintained. According to Fig. 7, in the paper, some additional ventilation was needed in a few cases to reach the eye irritation maximum density and considerably more ventilation in all cases was needed to reach the *commercially acceptable* density. In problems where recirculation is indicated, satisfactory means are available for removing the smoke particles. The pyridine vapors can be removed by adsorption by activated carbon as shown by tests made in the laboratory with a duct in which an electrostatic precipitator was installed and a bank of activated carbon canisters was located downstream. When the precipitator was in service and the canisters were removed from the duct the smoke was completely arrested but the odor came through with little, if any, noticeable reduction. When the precipitator was turned off and the activated carbon canisters placed in the line the smoke came through in volume but the odor concentration was very low. When precipitator and canisters were in service a very satisfactory job was done in removing both smoke and odor. The adsorption properties of pyridine by activated carbon are not far removed from those of benzene, a substance which resembles pyridine in chemical structure. It should be stated that, in the majority of problems which have been reviewed involving tobacco cases with insufficient ventilation, the owner emphasizes the eye irritation condition more often than the haze.

The author reports the same occupancy for basketball nights as for boxing nights. That is a little surprising because the entire central arena is occupied on boxing nights and very little, if any, of it on basketball nights. Probably, in both cases the Garden was not filled to capacity. Regardless of the reason for equal attendance, one thing is certain. On the boxing night, there was much smoking directly under the light beam of which there was none on the basketball night. This might have accounted for the lower optical transparency at the beam level as recorded in Fig. 6 and compared with Fig. 5. Will the author state, in this connection, if he observed a tendency for new smoke clouds to linger over the heads of patrons seated in the arena? In the not too distant past, inability to obtain a clear view of the ring has been an objection on the part of spectators in that section. The air distribution may, however, have been improved since.

It is hoped that the author will continue his studies along the simple and ingenious lines described. As he has said, the data secured cannot be safely extended to other types of structures. The ventilation used, in terms of cubic feet per minute per occupant, is lower than any values that have seemed proper for tobacco smoking conditions. Correspondingly, the resulting smoke densities leave something to be desired. It will be interesting if the author, in his future work on this subject, will include some smaller structures and in so doing point out what differences there may be in the amount of ventilation required on a large structure like Madison Square Garden, and that required on a structure of ordinary dimensions, for equal results, all other conditions such as space per occupant and cubic feet per minute per occupant being substantially the same.

C. A. ATHERTON[8] Louisville, Ky. (WRITTEN): In general I agree with the author's discussion of visual effects. I believe, however, that the introduction of the Weber-Fechner Law[9] is unnecessary and confusing. The effect on vision of the surrounding glare from the dispersal of light by the smoke is much more important than is the variable brightness of the objects viewed; from an illuminating engineering point of view this veiling glare is very important indeed. It is a problem similar in nature to that of trying to see through a fog by the light of automobile head lamps. It is interesting to note that the light is polarized, but probably no practical advantage can be taken of this fact.

The variation of visibility due to the background brightness; the fact that from some directions vision is by silhouette; and the greater weight which the author places on the distracting annoyance of the bright cloud of smoke rather than on variations in visibility; these are the sort of things an illuminating engineer would like to see investigated further.

It is unfortunate that only one condition of lighting was considered. Undoubtedly the lighting in the Garden is excellent, but I wonder if it has been investigated from the point of view of visibility through clouds of illuminated smoke. The author's field is that of getting rid of the smoke. Mine is that of producing lighting for good visibility. We look at every problem from two different points of view, but agree nevertheless, in a large degree, that at least in our two fields of engineering, there must be cooperation in order to arrive at the best and most economic solution. I congratulate the author upon bringing the two together.

As an illuminating engineer I note immediately that although the *commercially acceptable* transmission factor under the circumstances of the test was agreed to be 0.80, the transmissions actually observed during the game and match were considerably below that. This has a very familiar ring to the lighting man who talks of 50 or 100 foot-candles as his *commercially acceptable* intensities, but finds most installations giving 10-15 footcandles of intensity. It is curious how similar all such problems are in their fundamentals. So often we know how to do things much better, but we do not know how to get the people to adopt these better methods.

G. W. PENNEY, East Pittsburgh, Pa. (WRITTEN): The author is to be commended for starting a new type of investigation and for quickly publishing information to give engineers data on a subject on which nothing has been available. The quantitative increase in smoking between a basket ball game, the preliminary bout and the main bout is very interesting and useful.

I hope that the author is successful in stimulating further work by the Society. He is suggesting a new technical advisory committee to promote more extensive work, and I hope that this suggestion is carried out.

As the author suggests, a large number of additional tests are needed to obtain similar data on other sports events, night clubs, restaurants, etc. Madison Square Garden provided an excellent place for this particular method of testing because the space is large so that the light beam can be long, giving a large percentage of light absorption. However, we may be able to reflect a light beam several times across a smaller room so as to get a reasonable light absorption and adapt this method to smaller rooms.

These tests were based on using outside air which was sufficiently clean to give a negligible light absorption. However, in a large number of cities there are many *smoggy* periods when the outside air is so polluted with soft coal smoke as to give poor visibility in large rooms, even with all outside air and no tobacco smoke. In more extensive investigations, therefore, the cleanliness of the outside air must be considered.

As the author has suggested, another important subject is the purification of smoke-laden air so that it can be recirculated to permit maintaining good atmospheric con-

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[8] Reynolds Metals Co., Parts Div. (Member of *I.E.S.*).
[9] Loc. Cit. See footnotes 3 and 4.

ditions in auditoriums and similar buildings without the excessive heating and cooling that is required if all outside air is used for ventilation.

C. M. ASHLEY, Syracuse, N. Y.: This is a most interesting exploratory study in a practically new field. I do not believe that the result can be fully evaluated without asking Mr. Leopold whether he was smoking his pipe or not.

I would like to suggest one phase of the problem in which all of us can take a somewhat personal interest and that is the question of evaluating the number and activities of the smokers. That is one phase of the problem which is open to rather general observation and on which apparently, according to the observations made by Mr. Leopold, rather erroneous impressions have been gained.

Furthermore, I think that in smaller places the light problem will be sufficiently less acute so that we can concentrate our primary attention on the problem of eye irritation and on odor, and then correlate back with the results obtained by the methods suggested by the author to obtain a quantitative measure of contamination.

H. C. MURPHY, Louisville, Ky.: This paper is a valuable pioneering contribution to the study of air hygiene. Our Society has through the years been fortunate in having among its members men who were not afraid of new concepts. Twenty years ago this month, the author and his brother, Dr. Simon Leopold, outlined a valuable and frequently effective procedure for the relief of seasonal hay fever, bronchial asthma and similar allergic disorders by the use of air cleaners. In the ensuing years this procedure of creating spaces relatively free from air-borne allergic bacteria, spores and other air impurities has received intensive study from our Society and has made possible many economies in manufacturing processes. Perhaps because the economies were not so immediately evident, we have, in my opinion, up to this time failed to apply these principles adequately to the ventilation of workshops and places of public assemblage. We can measure with some degree of accuracy the effectiveness of electronic air cleaners in the removal of particulate matter, smoke and fumes. However, it is my belief that there is not available at the present time adequate data as to the effectiveness of air cleaners in removing all of the odor-producing and eye-irritating substances from an airstream. I believe that the study of odor control is greatly handicapped by the absence of adequate measurement standards and procedures. I support the hope that our Society will intensify its study and cooperation with other groups interested in this phase of air cleaning.

L. T. AVERY, Cleveland, Ohio: I want to thank Mr. Leopold for starting to give us a measuring instrument for smoke which can be used not only in an auditorium like this but in other smoky atmospheres. However, we have now no method of measuring oil mist. In a wide building there appears to be a lot more mist than in a narrow building. That is purely an optical illusion.

If we had such an instrument for measuring the smoke in a foundry it would be of great help. Actually, a dust count has nothing to do with it. You would be surprised to know that the dust count on several similar samples from a smoky atmosphere varied all over the lot. We must have some means of measuring smoke if we are going to incorporate it in smoke limitation codes.

I would suggest that the author's measuring devices be completed quickly and standardized to give us something with which we can measure.

When you are diluting air and mention 13 cfm per person, and a certain level of smoke, it must be pointed out it is for the method of air distribution. Shown in Fig. 4 of the paper. The fresh air is actually introduced above the smoker. All of the good air is mixed with the smoky air and the mixture driven down through the people. An infinite quantity of fresh air can be handled and still the smoke going through the audience will odorize the clothing because that is the way most of that smoke is going to escape.

On the other hand, if this system were reversed and if the clean air were introduced through the audience and the smoke were taken out of the top, the smoke and odor would be reduced and the contamination of clothes would be nil. That introduces

a problem in air conditioning. We all admit the value of recirculating air. So in assuming that we must have 13 cfm per person, let's emphasize the fact that it is with this method of air distribution.

Again, my corollary: I have seen in a foundry the fresh air brought in the top and blown downward through the smoke which had naturally accumulated at the top, with the result that the smoke was blown right back on the worker. That was a very expensive, elaborate system of heating and ventilating which was offered and sold in good faith.

AUTHOR'S CLOSURE: Mr. Munkelt's discussion mentioned the question of the distribution of seats near the ring. About 600 seats are involved. As you noted from Fig. 4, the side wall distribution would represent a fair sample of well over half of the seats in the arena, if you consider the induction effect. Undoubtedly he is basically correct in stating that it was a contributing factor. I think, however, a larger factor would be the character of the audience. Basketball is attended mostly by college men and women. There appeared to be many more pipes than there were cigars.

Mr. Atherton's question of the Weber-Fechner Law is in order. I wish to add, however, that in checking the literature I found that the electrical engineering handbooks, Standard, Pender and Del Mar, Eshbach's *Handbook of Engineering Fundamentals,* all accept the Weber-Fechner Law without question. Luckiesh frequently refers to the Weber Law; Parry Moon does not agree with any of them, and I cannot quite agree with Mr. Moon's argument.

In giving my paper, I have deliberately tried to avoid the terms of the illuminating engineers. Their data are in very difficult form to apply to a complex problem. In most of their data, as in most good scientific data, all other effects stand still where you vary one. That was not the kind of problem we had in Madison Square Garden.

In lighting the Garden, the possibilities of making changes, I think, are great. There may be, for example, the question as to whether the background would not be better white instead of a dark color. I mentioned in my paper that no experiments were made in the lighting. The existing lighting was accepted as it was.

I agree with Mr. Penney's remarks on outdoor smoke. I have known cases in Pittsburgh where it was necessary to close the fresh air intake to a theater and proceed with 100 per cent recirculation in order to get rid of smoke.

1278

No. 1278

NOISE RATINGS OF VENTILATING FANS

By W. H. Hoppmann, II* and Fred Lager,** New York, N. Y.

In VENTILATING systems, noise generated by fans is generally undesirable, but especially so in the case of Naval Vessels. Accordingly, in 1939 the Navy Department, in connection with its work to reduce noise levels on naval vessels, formulated a noise specification for ventilating fans. The testing technique described in the specification was subsequently adopted by the *National Association of Fan Manufacturers* and in 1942 it was incorporated in their code entitled Sound Measurement Test Code for Centrifugal and Axial Fans.

Both in the *N.A.F.M.* Code and in the Navy Specification, the noise rating of the fan is defined as the arithmetical average of the sound levels at seven positions around the fan, as shown on Fig. 1. The position of the microphone at each station is required to be in a plane parallel to the floor and coinciding with the horizontal centerline of the fan.

Some investigators have considered that sound power output (SPO) would constitute a more satisfactory rating for ventilating fans. There are a number of points in favor of the use of this method of rating. It can serve for acoustical design as well as for specification purposes. It provides a more refined rating which is independent of the distances of the microphone from the fan, whereas the present method gives an average decibel rating which depends on the specific configuration of microphone positions used to determine it. However, it may also be useful to have a noise rating in a logarithmic scale, such as the decibel, because the ear response is nearly logarithmic.

An average decibel rating could be obtained from the SPO by computing for a chosen distance, the sound level which, if uniform over a hemisphere of radius equal to that distance, would give the acoustical power of the fan.

The test method used by the Navy and the *N.A.F.M.* does not differ essentially from the test method required in the determination of the SPO. In both cases sound levels are determined with standard meters. The difference really lies in the method of calculation and possibly in the number and locations of microphone positions. In the case of the present specified method the sound levels are simply averaged arithmetically while in the SPO method the levels are operated on to evaluate the power.

A method of estimating sound power output is given as follows:[1] If a piece of equipment is radiating sound into an infinite medium, that is, into a medium in which no reflecting surfaces are present, the total output from the

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[1] Applied Acoustics by Olson and Massa.

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equipment can be obtained by measuring sound levels at a reasonably large distance from the source and at various points over a spherical surface of which the sound source is the center. The total power output will then be equal to the integrated components of the equivalent energy flux over the surface of a sphere.

For the case of a ventilating fan mounted on a hard floor, a hemisphere may be used on the assumption that all of the sound directed to the floor is reflected through the surface of the hemisphere. The hemispherical surface should be divided into suitable areas and the sound level measured in each area.

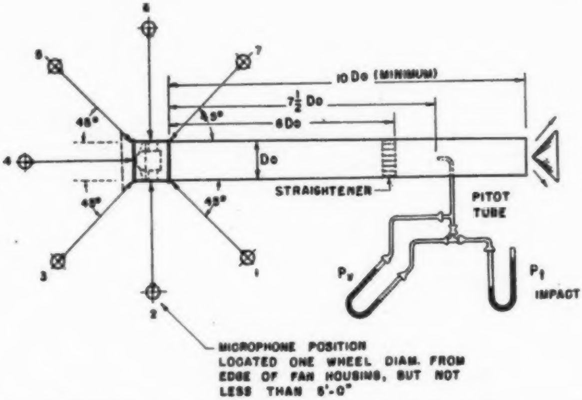


FIG. 1. FAN TEST ARRANGEMENT USED BY THE NAVY DEPARTMENT AND THE N.A.F.M.

The SPO may then be obtained by a numerical integration or summation of the products of the power per unit area and area corresponding to each of the various sound level measurements.

A simple tabular form for the purpose of routine testing and rating can be set up so that sound level readings in decibels, ambient sound levels and acoustical power per unit area computed from the sound levels may be recorded. After the elementary calculations to convert the decibel readings into power per unit area are performed and allowances for ambient are made, the SPO may be readily computed.

It should be noted that if the ambient sound level is less than 10 decibels below the readings obtained with the fan operating, it is considered necessary to make an allowance for it.

It is not practicable at the present time to measure sound levels in the airstream in a duct connected to a ventilating fan, mainly because of the lack of a suitable device to measure reliably this portion of the radiated sound. However, since the present fan testing codes are mainly designed to evaluate fans on a relative basis with regard to noise, this point is not too serious. For

more precise analysis and rating, provisions should be made to estimate the sound that is radiated into the duct.

It should be clear that for either method of evaluating the acoustical performances of a ventilating fan, a space approximating a free field is required. Since an infinite space with no reflecting surfaces is just an ideal concept and the use of quiet out-of-doors as an approximation is impracticable because of weather and other conditions, it is highly desirable to provide a soft standard room for test purposes. This room is essential for the SPO method and, in the case of the method prescribed in Navy specification and *N.A.F.M.* Code, would serve to eliminate controversies about results obtained in different test spaces.

On the basis of the experience that the Navy has gained in the testing of ventilation fans for noise characteristics, it has now designed and built a special fan test room to which the remainder of this paper will be devoted.

The primary factors influencing the design, and particularly the size, of the fan test room, were the Laboratory space available and the cost of construction. On the one hand, the size and shape of the fan test room were governed by the 20 ft spacings of the concrete columns of the Laboratory building and, on the other hand, it was necessary that the room be at least large enough to encompass the largest test fan with sufficient room for microphone measurements. In addition the room had to be isolated from vibrations of the floor on which the room is set, acoustically treated with sound absorbing material to reduce the sound wave reflections in the room and insulated to reduce the transmission of external noise into the room. The acoustical insulation, however, which must be capable of insuring ambient noise levels of 68 db or lower in the room, had to be maintained in spite of the fact that the nature of the test requires several relatively large openings in the room; a large low velocity, air intake and a smaller opening for the discharge duct from the test fan. A door was also provided for access. With indoor Laboratory space available for the construction of the fan test room, weatherproofing was not a consideration in the design and the room as constructed by the Navy is not intended for outdoor construction.

The maximum interior dimensions of the fan test room are 20 ft wide by 20 ft long by 10 ft high, exclusive of a 3 ft high by 20 ft wide air intake in the upper portion of the structure, as shown in Fig. 2. This provides sufficient space to test the largest Navy ventilating fans, including space required for microphone measurements. The irregular hexagonal shape of the room was governed largely by the 20 ft center to center spacings of the concrete columns of the laboratory building in which the fan test room is located. The asymmetry of the structure, however, is also of some value in reducing the standing wave pattern that occurs in the room during a test of a fan.

Wood construction was used for the fan test room in preference to steel, brick or concrete, because of the ease of construction, reduced weight, and lower cost. Heavy wood framing, covered externally with two layers of tongue and groove sheathing, separated by double thickness of heavy asphaltic paper was used for the walls and roof. The upper surface of the intermediate ceiling, the underside of the roof and the floor, were similarly constructed. Floor joists, roof beams, ceiling beams, ceiling girders and girder posts were also constructed of wood. Fig. 3 shows a photograph of the exterior of the fan test room.

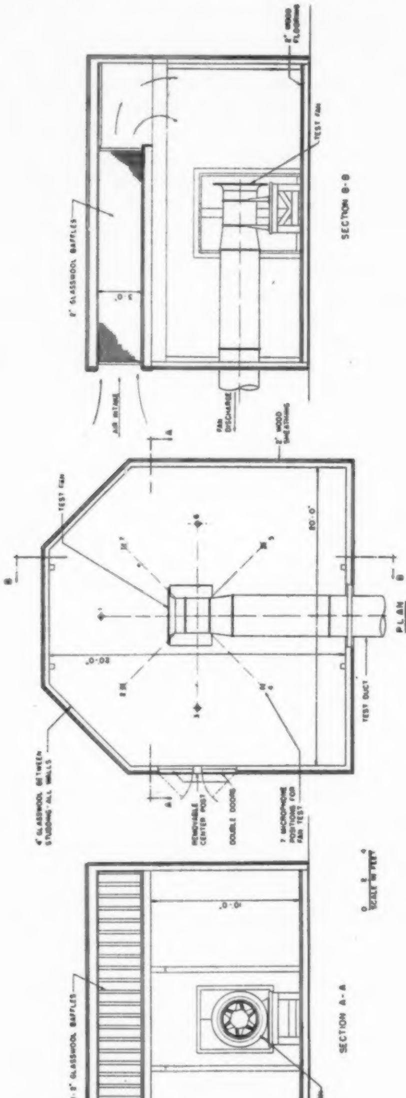


FIG. 2. PLAN AND SECTIONAL ELEVATIONS OF FAN TEST ROOM

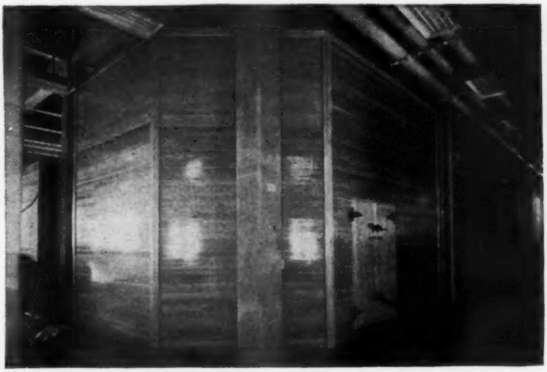


FIG. 3. EXTERIOR VIEW OF REAR AND SIDE OF FAN TEST ROOM

Since the boundary surfaces of the test room greatly affect the sound level in the room due to the number of reflections of the sound waves from these surfaces, the interior of the walls and the underside of the ceiling were acoustically treated by covering them with 4 in. thick fiberglass over which were spread two layers of muslin and wire mesh. The glass wool in blankets was placed between the 4 in. studding of the walls and between the ceiling joists. No acoustical treatment was applied to the floor. A photograph of the interior of the room is shown on Fig. 4.

A low ambient noise level is important in a room for noise testing. In order to accomplish this for the fan test room, with the type of wood construc-

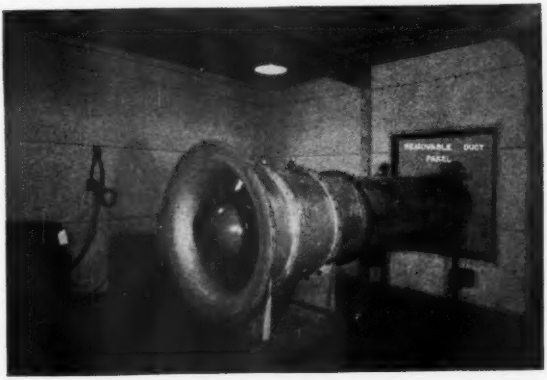


FIG. 4. INTERIOR OF FAN TEST ROOM LOOKING TOWARD REMOVABLE DUCT PANEL

FIG. 5. EXTERIOR VIEW OF FAN TEST ROOM SHOWING AIR INTAKE AND REMOVABLE DUCT PANEL

tion used, the two layers of tongued and grooved wall and roof sheathing were tightly fitted and end matched, with the joints staggered. The two ply asphaltic paper used between layers of wood sheathing were continuously lapped around the structure in order to keep air leakage and accompanying airborne noise at a minimum. Wood draft stopping and caulking were used at the intersection of walls, ceiling and roof, to eliminate the possibility of airborne noise entering the room. However, a large air intake is essential for fan testing and it was necessary to treat this intake acoustically to absorb the

FIG. 6. INTERIOR OF FAN TEST ROOM LOOKING TOWARD DISCHARGE OF AIR INTAKE

noise coming in from outside of the room and insure a low ambient noise level within. The air intake to the room is a 3 ft high by 20 ft wide channel containing 28 baffles, approximately 14 ft long and 3 ft high, of wood frames containing 2 in. of glass wool, covered both sides by muslin and wire mesh. The space between adjoining baffles is 6 in. clear and the free area of the air intake is approximately 72 per cent. Photographs of the air intake are shown on Figs. 5 and 6. The discharge of the air intake in the room is shown on Fig. 7.

The other two openings in the fan test room, one for the discharge duct and one for access to the room, were insulated against noise transmission from the

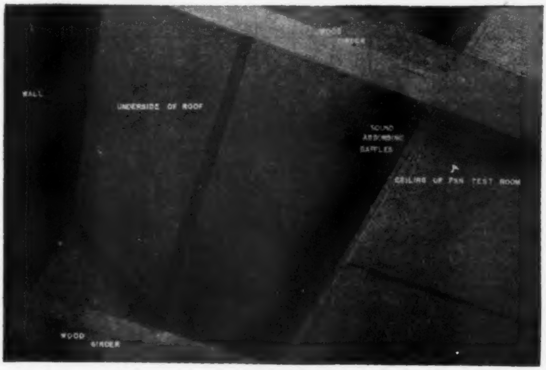


FIG. 7. INTERIOR OF ROOM SHOWING DISCHARGE OF AIR INTAKE (LOOKING UPWARD)

outside, by providing means for closing the openings during a fan test. The discharge duct opening in the room is a steel framed opening provided with several pairs of 2 in. thick, tongued and grooved, removable panels, each pair of which is cut to fit a different size of circular test duct. The pair of panels shown on Figs. 4 and 5, is bolted to a felt lined angle iron frame after the test duct is in position. The entrance to the room is through a pair of 2 in. thick wood doors, provided with dog clamps which, when closed, force the doors against a continuous wood and piano felt doorstop and gasket and so provide reasonably airtight and soundproof doors. The center post of the door frame is removable for the access of the large fans.

The fan test room is provided with electric lights and power outlets on the walls, which were so made that there is no rigid conduit connection between the room and the power source. Where it was necessary to pierce the walls, floor, ceiling or roof, flexible conduit was used and the spaces around the conduit were plugged and caulked.

Isolation of the fan test room from vibrations of the foundations on which it is built is essential to insure both low ambient noise levels in the room and reliability of the test results. One inch hairfelt, placed between the concrete

floor of the laboratory building and all bearing surfaces of the Navy room, was satisfactory. However coiled springs or commercial types of vibration eliminators would have to be used, if the floor vibrations were of such amplitude and frequency as to make the 1 in. hairfelt ineffective.

The determination of the sound wave pattern or the presence of standing waves in the room at frequencies predominant in ventilation fans would indi-

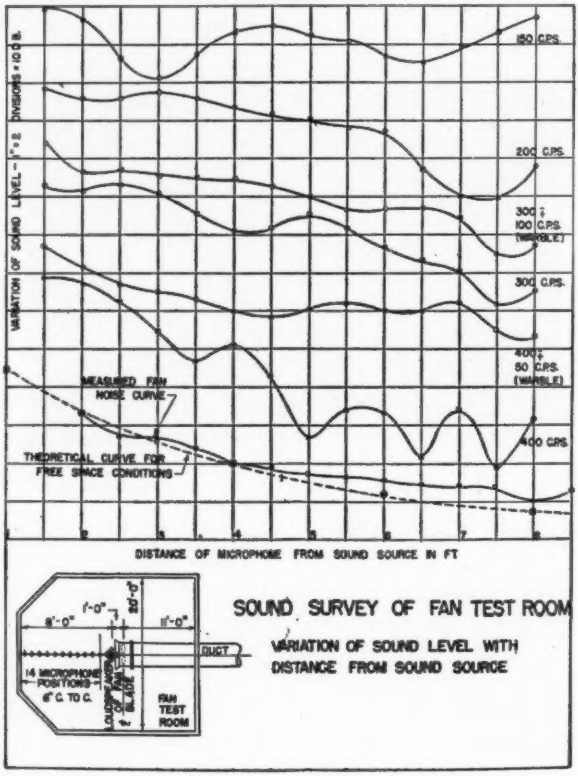


FIG. 8. SOUND SURVEY OF FAN TEST ROOM

cate the acoustical value of the room for fan testing. Fig. 8 shows the results of a sound survey at various distances from a non-directional loudspeaker unit on which was impressed both warbled and pure frequencies. Also shown on Fig. 8 is the curve of variation of sound level of a fan with microphone distance from the fan. For comparison, there is superimposed on this curve the theoretical curve for free field conditions, drawn through the point representing the microphone position closest to the fan. It should be noted that part

of the deviation is attributable to the fact that the fan is not a point source, while the theoretical curve for free field conditions is based on a point source. The most pronounced wave pattern was obtained with the single frequency sounds as compared to the warbled frequencies and the many frequencies of the fan. This would be expected if certain of these single frequencies were identical with the natural frequencies of the air mass of the room. In general

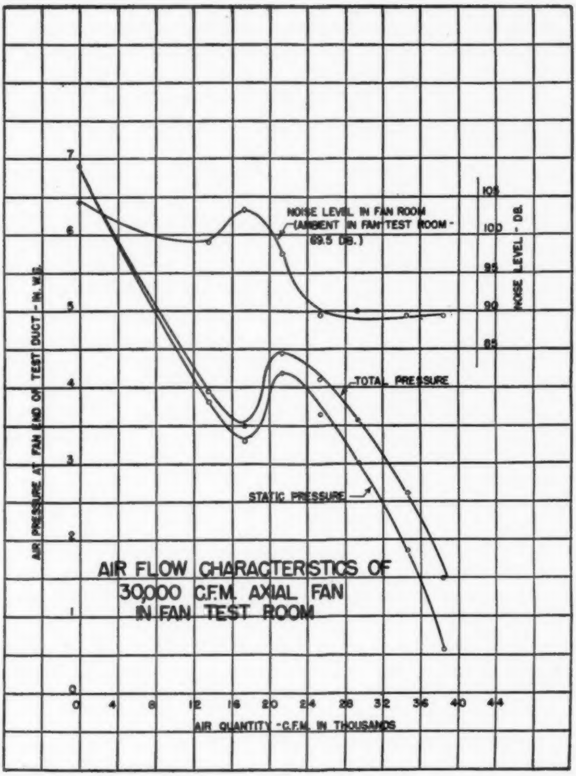


FIG. 9. AIR FLOW CHARACTERISTICS OF 30,000 CFM AXIAL FAN IN FAN TEST ROOM

the standing wave pattern in the room is of small amplitude so that it will not introduce any appreciable error in the noise measurements in the fan test room.

The determination of the sound transmission loss through the room would indicate the effectiveness of the room in attenuating external noise and the ambient noise levels that could be expected in the room. This was done, with a fan and duct in test position in the room, by generating a high uniform noise

level of 89 db outside of the room and measuring the noise levels inside the room. It was found that the room attenuated 14 db and that an ambient noise level of 68 db can be obtained in the room when the Laboratory is at normal operation.

The noise generated by the fan will vary with the air flow pattern in the fan test room and the air flow pattern will vary, depending on the location of the fan in the room. Tests were conducted to determine both the effect of the fan location on the noise generated by the fan and the best position of the fan for test. By smoke flow tests, air flow measurements and by determining the noise performance curve of fans at many locations in the room, it was found that the most stable air flow condition, least turbulence and best performance were obtained with the fans in the center of the room. Axial fans were found to be much more sensitive to the air flow pattern in the room than were centrifugal fans.

The original room built by the Navy was provided with an intake located at the end of the room opposite the fan duct discharge. The *S* shaped path of the air from outside the room to the fan inlet resulted in air turbulence and unreasonably high noise levels generated by the fan. The present modified room now has the air intake at the same end of the room as the discharge duct, the air following a *U* shaped path from outside the room to the fan inlet, resulting in more stable air flow conditions in the room and subsequent lower noise levels generated by the fan. At the discharge of the air intake to the room, many arrangements of splitters, vanes, deflectors and splash plates were tested to determine the effect on the noise generated by the fan. Although the use of vanes and splitters resulted in a better distribution of the air across the discharge of the air intake, no improvement in fan performance was observed and the additional expense of a permanent installation did not seem warranted.

Both centrifugal fans and axial fans up to 30,000 cfm have been tested in the fan test room, for both airflow performance and for noise. The data obtained appeared to be a valid measure of the merits of the individual fans. Typical performance curves are shown on Fig. 9 and were obtained in testing a 30,000 cfm axial fan. The maximum pressure loss across the room was 0.145 in. water at 38,000 cfm for the axial fan rated at 30,000 cfm at 3.0 in. water.

The largest fan tested in the fan test room was the 30,000 cfm axial fan, 45 in. in diameter, and 52 in. long, exclusive of the inlet bell mouth, and the room provided ample space for the seven microphone positions specified in the Code.

One of the most important advantages of the fan test room for fan testing is that it is reproducible and allows for standardization of the test space. In the past spaces for testing fans for noise have been defined only in general terms with a general requirement that they approximate free field. However, fan testers, while conforming to the specification in this respect, may obtain different results with the same fan tested in different spaces due to space effects. It is highly impracticable to endeavor to compute or correct for these effects. The fan test room is an attempt to eliminate controversial space effects encountered in fan testing and to provide a means of further standardization of fan test methods.

Acknowledgment is made to Capt. T. H. Urdahl, USNR, for his sponsorship of the ventilating fan testing room for the Navy Department; to Lieut.

Comdr. H. M. Gurin, USNR, for his constructive criticism and encouragement during the work; to John Lind, Principal Materials Engineer, Material Laboratory, New York Navy Yard, under whom the test room for the noise rating of fans was designed and built; and to the authors' associates in the Material Laboratory, New York Navy Yard, for considerable assistance in conducting tests to determine the acoustical properties of the room and the performance of fans tested in the room.

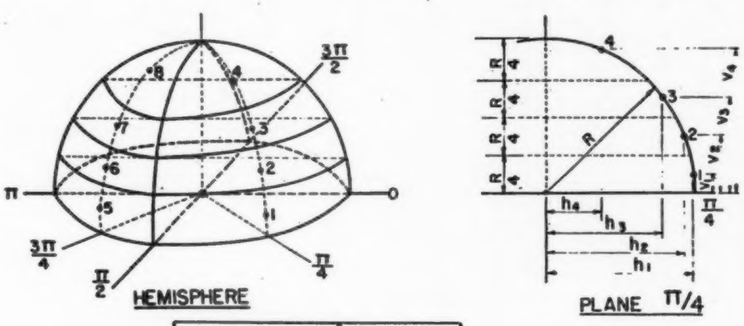
The opinions or assertions contained herein are the privates ones of the writers and are not to be construed as official or reflecting the views of the Navy Department, or the Naval Service at large.

APPENDIX

The interest shown in the *sound power output* method of rating ventilating fans or other noise producing equipment has prompted the authors to reproduce the slides, which are shown in Figs. A, B and C.

Fig. A shows the theoretical hemisphere of radius R, in which the sound source is located at the center. The surface of the hemisphere has been divided into 16 equal areas by cutting the hemisphere with 3 planes of latitude, R/4 apart, by 2 planes of longitude, 180 deg apart and cutting the hemisphere at 0, $\frac{\pi}{2}$, π, and $\frac{3\pi}{2}$. The centers of the areas formed by the intersections of the planes on the hemisphere, stations 1 to 16, lie in longitudinal planes, $\frac{\pi}{4}$, $\frac{3\pi}{4}$, $\frac{5\pi}{4}$, and $\frac{7\pi}{4}$.

Considering any one of the planes, *i.e.*, plane $\frac{\pi}{4}$, it contains stations 1, 2, 3, 4, the centers of four areas of the spherical surface. The location in space of stations 1, 2, 3, 4, is described by horizontal distances h, from the center of the sound source and



MICROPHONE STATION				LOCATION	
$\frac{\pi}{4}$	$\frac{3\pi}{4}$	$\frac{5\pi}{4}$	$\frac{7\pi}{4}$	h	v
1	5	9	13	.992 R	.125 R
2	6	10	14	.927 R	.375 R
3	7	11	15	.781 R	.625 R
4	8	12	16	.484 R	.875 R

FIG. A. LOCATION OF 16 MICROPHONE POSITIONS FOR SOUND POWER OUTPUT (SPO)

E = POWER OUTPUT/CM^2 IN MICROWATTS/CM^2

I_0 = 10^{-16} WATTS/CM^2 = 10^{-10} MICROWATTS/CM^2 = REFERENCE LEVEL

I_R : POWER OUTPUT/CM^2 CORRESPONDING TO db_r

I_A : POWER OUTPUT/CM^2 CORRESPONDING TO db_a

dA = DIFFERENTIAL SURFACE AREA

A = SURFACE AREA OF EACH ELEMENT IN CM^2 = $\frac{\text{TOTAL AREA OF HEMISPHERE}}{16}$

db_r = SOUND LEVEL IN db AT MICROPHONE OF TEST EQUIPMENT PLUS AMBIENT

db_a = AMBIENT SOUND LEVEL IN db

TOTAL SOUND POWER OUTPUT (SPO) OF TEST EQUIPMENT = $P = \int_{\text{SURFACE}} E\,dA$

IF SURFACE IS DIVIDED INTO 16 EQUAL AREAS THEN $P = A\sum_{i=1}^{16} E_i$ (APPROX.) (1)

$$E = (I_R - I_A)$$

$$\text{BUT} \quad I_R = I_0\, 10^{\frac{db_r}{10}} \quad \text{AND} \quad I_A = I_0\, 10^{\frac{db_a}{10}}$$

$$\therefore E = (I_0\, 10^{\frac{db_r}{10}}) - (I_0\, 10^{\frac{db_a}{10}})$$

$$\text{OR} \quad E = I_0 (10^{\frac{db_r}{10}} - 10^{\frac{db_a}{10}}) \quad (2)$$

SUBSTITUTING EQ. (2) INTO EQ. (1)

$$SPO = (10^{-10})(A)\sum_{i=1}^{16}(10^{\frac{db_r}{10}} - 10^{\frac{db_a}{10}}) \text{ MICROWATTS}$$

FIG. B. CALCULATION OF SOUND POWER OUTPUT

1	2	3	4	5	6
	db_a	db_r	$10^{\frac{db_a}{10}}$	$10^{\frac{db_r}{10}}$	$(10^{\frac{db_r}{10}} - 10^{\frac{db_a}{10}})$
1	69	83	7.9 X10^6	199 X10^6	191 X10^6
2	69	83	7.9 X10^6	199 X10^6	191 X10^6
3	70	85	10.0 X10^6	316 X10^6	306 X10^6
4	69	84	7.9 X10^6	251 X10^6	243 X10^6
5	70	86	10.0 X10^6	398 X10^6	388 X10^6
6	71	85	12.6 X10^6	316 X10^6	303 X10^6
7	69	87	7.9 X10^6	501 X10^6	493 X10^6
8	69	85	7.9 X10^6	316 X10^6	308 X10^6
9	69	85	7.9 X10^6	316 X10^6	308 X10^6
10	69	83	7.9 X10^6	199 X10^6	191 X10^6
11	70	84	10.0 X10^6	251 X10^6	241 X10^6
12	69	83	7.9 X10^6	199 X10^6	191 X10^6
13	70	85	10.0 X10^6	316 X10^6	306 X10^6
14	70	87	10.0 X10^6	501 X10^6	491 X10^6
15	69	87	7.9 X10^6	501 X10^6	493 X10^6
16	69	85	7.9 X10^6	316 X10^6	308 X10^6

$$\sum_{i=1}^{16} = 4{,}952 \times 10^6$$

ASSUME RADIUS OF HEMISPHERE = 5 FT. = 152.4 CM

AREA OF ONE ELEMENT = $A = \frac{4\pi R^2}{2 \times 16}$

$A = 8{,}800$ CM.^{2}

$I_0 = 10^{-10}$ MICROWATTS/CM^2

$$P = I_0 A \sum_{i=1}^{16}(10^{\frac{db_r}{10}} - 10^{\frac{db_a}{10}}) = 10^{-10}(8{,}800)(4{,}952 \times 10^6) = 4{,}358 \text{ MICROWATTS (SPO)}$$

THE AVERAGE db READING CORRESPONDING TO THIS SPO IS:

$$db = 10 \text{ LOG } \frac{I}{I_0} = 10 \text{ LOG } \frac{\frac{4{,}358}{(8{,}800)(16)(10^6)}}{10^{-16}} = 10 \text{ LOG}(2.9 \times 10^8) = \underline{84.6 \text{ db}}$$

FIG. C. TABLE OF CALCULATIONS FOR SOUND POWER OUTPUT (SPO)

by vertical distances V, above the sound source. The table shown in Fig. A locates all the microphone stations in terms of any radius of sphere R. It should be noted that the 16 microphone stations are described by four sets of coordinates, in four longitudinal planes.

From a practical standpoint, these microphone stations can be covered by using a microphone on an adjustable stand. For example, at one microphone setting equal to V_4, four noise readings can be taken, one each at stations 4, 8, 12 and 16, each at a horizontal distance h_4 from the center of the sound source, but each in a different longitudinal plane, $\frac{\pi}{4}$, $\frac{3\pi}{4}$, $\frac{5\pi}{4}$ and $\frac{7\pi}{4}$. In a similar manner, the remaining stations can be covered.

Fig. B shows the fundamental calculation of the sound power output of a noise source if the surface of hemisphere is divided into 16 equal areas. The SPO can be calculated by summing the products of the power per unit area and the area of the corresponding element of the spherical surface. These calculations have been reduced to a simplified tabular form shown on Fig. C. Microphone station, ambient sound level in db, and sound level in db of test equipment plus ambient, are recorded in columns 1, 2 and 3. Column 4 and 5 of the table are for numerical quantities required in the equation of SPO and column 6 is the difference between columns 4 and 5. The 16 values in column 6 are summed and the total is substituted in the equation of SPO. The values of I_o and A in the equation are noted in Fig. C.

The SPO value so obtained can be converted to a corresponding average db reading as shown on the bottom of Fig. C. The SPO is divided by 10^6 to convert it to watts, and by the total area in cm^2 to convert it to watts per cm^2. If this value is then divided by the reference level in watts per cm^2 and the logarithm of the result multiplied by 10, the average db value corresponding to the SPO, is obtained.

DISCUSSION

R. D. MADISON, Buffalo, N. Y. (WRITTEN): This paper emphasizes the value of standardization of fans and noise ratings conducted by the Navy Department. It also mentions the close cooperation existing between the Navy and the fan manufacturer.

You may be interested to know what the Committee on Research is doing in respect to sound testing and sound test codes. Members of the Society are interested in two somewhat different types of fan installations, those with ducts and those without. For this latter case (unitary equipment) the SPO method is particularly applicable. Generally, the units are not large and we are interested in the total energy and noise radiated from the unit. In the case of fans with inlet and outlet ducts the case is somewhat different. To know the total energy or noise is not sufficient unless it is separated into the two principal components (1) that radiated into the fan room and (2) that transmitted out through the inlet and outlet ducts. For this reason it is our plan to investigate both methods of test. As soon as personnel is available at the new Research Laboratory this project will get under way.

The chief criticism of the Navy and N.A.F.M. sound test codes is that they do not represent all types of fan installations. For example, a fan with both inlet and discharge ducts would have less sound radiated into the equipment room than one with discharge duct only. However, it would be doubtful if there would be much change in the amount communicated through the discharge duct. Probably the reduction in the amount radiated into the fan room now would be transmitted back through the inlet duct. Tests which we hope to make should throw some light on this point.

It is my opinion that a close correlation will result between the sound recorded by the method of test as outlined in this paper and the SPO method.

E. R. QUEER, Washington, D. C.: We are not depending only upon the tests that are made in the laboratory to determine the noise of these fans, but we must carry on to the ship to see that the fan is properly installed, so that the noise level does

not become excessive. Just recently we have completed several noise surveys on new combatant ships to determine wherein they had difficulties. There were a number of complaints of noisy ventilation.

The point that I wanted to emphasize particularly is that we are carrying on, in addition to this regular work that is being done in the laboratories and with the fan manufacturers, setting up specifications to limit the noise level, and seeing that the installations are properly made aboard ship.

C. M. ASHLEY, Syracuse, N. Y.: I would like to take this opportunity to call your attention particularly to the sound energy method described by Mr. Hoppmann. I believe that it fits into the general pattern of trying to get down as close as we can to fundamentals. Most of the previous methods used have been somewhat empirical in nature. By use of the sound energy method it is possible by dividing by the total absorptivity to get back to the intensity of the sound in a given space; and from the intensity one can convert into the sound level, a thing which has not been possible to do with good accuracy before. I believe it should be recognized as a very distinct step forward in the measurement and interpretation of the sound level of equipment.

T. H. URDAHL, Washington, D. C.: I wish to thank the authors on behalf of the Bureau of Ships for their preparation of this paper and its presentation here. I also wish to state the objective of the Bureau in making the initial assignment to the Material Laboratory of the New York Navy Yard for the design of the fan test room which you have just heard described. It has been our practice, as the authors of the paper have outlined, to type test manufacturers' fans for meeting the minimum requirements for building Navy fans. This has been a cumbersome procedure involving shipping the fan to the laboratory for the test and upon possible failure in the test, shipping it back to the manufacturer and reshipping it, after improvement, to the laboratory. Failures have largely been caused by the development of the fan in an environment quite different from that in which it was tested for noise.

It was our objective to try to develop at a low construction cost consistent with its purpose a test room, which could be uniformly used by any fan manufacturer, and which would be within the reasonable capacity of any fan manufacturer to build and own. Thus, the development of all fans could take place in an identical and standardized environment.

We hope in time to eliminate the necessity for shipping fans to New York. Fans will be developed by the manufacturer in his own plant in a sound room of this type, so that when type approval is requested it will only be necessary to send the necessary accredited personnel to the manufacturer's plant to witness the tests.

There is one further item of standardization to be accomplished in the matter of sound testing, namely, the calibration of the test instrument. Steps are under way at present to set up a standard calibration of sound measuring instruments, microphones, etc., at the U. S. Bureau of Standards.

The comment of the fan industry on this test room is solicited. I also wish to express my appreciation and that of the Bureau to the fan industry at this time for their extremely cooperative effort not only in producing the thousands of fans that the Navy needs, but in constantly cooperating with us in developing better fans for Naval vessels.

1279

No. 1279

PRINCIPLES OF ULTRAVIOLET DISINFECTION OF ENCLOSED SPACES†

By L. J. Buttolph,* Cleveland, Ohio

GERMS AND VIRUSES of the common respiratory diseases are now known to be in the air of occupied rooms along with dust, odors and carbon dioxide. The occupants themselves are the unconscious sources of most of this contamination. With yeast cells and mold spores, the bacteria and viruses either float free in air as a living dust or are themselves attached to inert dust particles, or, in both forms, may be at rest on the walls and floor. The extent of the air contamination is only partially obvious in a motion picture projection beam or in a similar beam of sunlight through a darkened room because the bacteria and viruses themselves are too small to be individually visible even under such conditions.

The original sources of air contamination by pathogenic bacteria and viruses are generally the noses and throats of room occupants but the walls and floors and ventilating ducts become secondary sources of air infection which must be taken into account under some conditions. The immediate aim of air disinfection is to reduce the concentration of living airborne pathogenic organisms at the breathing level by killing them and by dilution with air which is free of them.

A discussion of the probable mechanism of respiratory infection would be outside the subject of this paper if it were not essential for perspective as to the significance of the time element involved in the processes of contamination, disinfection and the accumulation of pathogenic organisms in the human respiratory system.

Nose and Throat as Air Sampler and Filter

An important function of the nose and throat is the removal from breathed air of dust particles which include not only bacterial organisms enclosed in droplets of water, but especially the so called *droplet nuclei* which are the particles of living dust left when the enclosing droplets of water, in which they may have been carried from noses and throats, have evaporated in space. As a consequence of this natural bacteria air filtering operation of the nose and throat, these cavities carry on their surfaces at all times a representative sampling of the dust and bacterial content of the air breathed by the individual for the many hours previous. For this reason every person is at times a carrier of pathogenic organisms awaiting only an opportunity, provided by mechanical lesions or the invasion of other sources of infection to make their own characteristic entrance into the underlying tissues as an active infection which may be diagnosed generally as streptococcic or virus, or, specifically, as bron-

† Based on a paper presented at the Annual Meeting of the *American Public Health Association*, New York City, October 3, 1944.

* Engineer, Nela Park Engineering Div., General Electric Co.

chitis or scarlet fever or measles or mumps, depending upon the organisms involved and the location of lesions.

Many of the organisms collected from air by the moist hairs and tissue folds in the nose and throat are, in turn, expelled from the nose and throat by coughing, sneezing and talking, as has been graphically demonstrated by photographic methods, to be again inhaled by another person. It is of special interest to note that even when the organisms are expelled in minute droplets of water these droplets are so small that most of them evaporate before coming to rest on a surface and the living dust particles float away to settle out of the air only after many hours.

Organisms which are not thus expelled from the nose and throat obviously do not remain there indefinitely, but are disposed of mechanically along with dust by the mucous secretions and the living organisms are, to some extent at least, destroyed by the natural protective and disinfecting action of the nasal and throat secretions. Thus we have a dynamic condition in which the actual concentration of living pathogenic and nonpathogenic organisms at any particular time may be represented as resulting from an equilibrium between the rate of collection from breathed in air and the rate of disposal by expelled air and the body processes.

It is obvious then that the concentration of germs in the nose and throat may be reduced by reducing their concentration in the breathed in air by air disinfection. Here again we have a dynamic condition in which the actual concentration of living organisms in the air is an equilibrium between the rate of contamination by the healthy *carriers* and the diseased sources of infection, on the one hand, and the rate of decontamination, or disinfection, by the normal death of the bacteria themselves, by the filtering action of noses and throats, by dilution with fresh air and by the bactericidal action of chemical vapors or ultraviolet energy.

TIME RATE OF COLLECTING BACTERIA

It is important to emphasize here that these processes are much slower than ordinarily realized, as may be illustrated by the fact that a person can walk through a room containing an extremely high concentration of dust without acquiring a noticeable deposit in the nose; while 6 to 8 hours in a room not visibly contaminated with dust will provide most obvious evidence of the effective dust filter the nose can be. The inference is that a person does not acquire a respiratory infection from a passing exposure to a sneeze but rather from a much longer exposure to such concentrations of respiratory disease organisms as are characteristic of enclosed and occupied spaces. It is possible that the seriousness of the sneeze or cough as a momentary hazard to a passing person may have been overemphasized.

Time, the rate of change, is also very important in connection with any process of decontaminating or disinfecting air because its effectiveness is always relative to the continuing contamination and, in effect, the rate of decontamination must be greater than the rate of infection to secure any improvement in the condition of the air and equal to it for maintenance of the improvement. A rigorous analysis of the condition existing where the processes of contamination and disinfection are taking place concurrently is ex-

tremely involved and is seldom attempted because of the difficulty of determining the infection rate from human beings and in terms of the air sampling of pathogenic organisms. The problem can be simplified somewhat by the use of a mechanical infecting device or *infectee* as has been reported by Wells (1)[1] and used by many others. As is the case with milk and water, a dangerous concentration of pathogenic organisms may be difficult to determine by the available methods of sampling and indices of disinfection are used in practical work. In the laboratory artificially produced concentrations of harmless B-coli much greater than ever occur naturally with pathogenic bac-

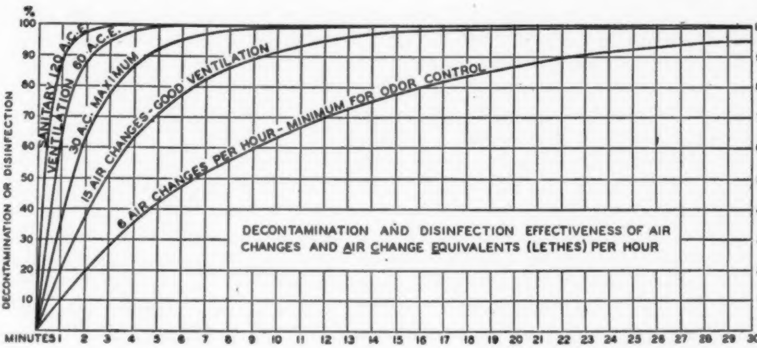


FIG. 1. THEORETICAL RATES OF AIR DECONTAMINATION IN ROOMS

teria, facilitate accurate determinations of the changes in concentration resulting from dilution or disinfection.

SUMMER AIR AND VENTILATION

Since there is as yet very little correlation between the occurrence of airborne transfer of respiratory diseases and directly determined concentrations of airborne organisms, it seems best to work backwards from those conditions of natural ventilation known by experience to insure a minimum transfer of respiratory infection. It seems to be the opinion of those who have studied the problem that a ventilation rate of 120 air changes per hour is essential to duplicate the conditions of summer living, with 60 air changes as a minimum in any case and with 240 changes desirable under some conditions. Fig. 1 has been prepared to relate these rates of air change to those provided by current ventilating practice.

SCHOOL VENTILATION AS AN EXAMPLE

Under average school housing conditions the 30 cfm of air per child and the *make-up air,* in the best heating and ventilating systems, provides about

[1] Numerals in parentheses refer to Bibliography.

6 air changes per hour. This is one air change in 10 min, Fig. 1. This is a ventilating condition such that in the course of 10 min there is brought into a school room, for example, with theoretically perfect and continuous mixing a total cubage of air equal to the volume of the room. It is known that as a result of this process of dilution there will remain at the end of the 10-min interval 36.8 per cent of the initial air, carbon dioxide, odor, dust and bacteria, a decontamination of 63.2 per cent. During the next 10 min there will be a further reduction of 63.2 per cent in the remaining 36.8 per cent of the initial bacteria, providing a disinfection at the end of 20 min of 86.5 per cent. As the process continues a practically complete disinfection of 99.99 per cent will only be reached after about 1½ hours of time and 9-10 air changes, but this, it is important to note, is under the condition that there is no continuing new contamination during the hour, in other words, the first hour after the children have gone home. In contrast with this ventilation which was originally specified for the disposal of carbon dioxide and odors it should be noted that the maximum practical by mechanical means of air circulation is 30 air changes per hour which would still require about 10 min to disinfect the air of a recently occupied room. In contrast with this maximum mechanical possibility and the fact that 15 air changes per hour represent de luxe ventilation, it should be noted that, since only the *make-up air* in any air conditioning system is effective in reducing the bacteria count, the air disinfection by dilution provided by current ventilation practice is much less than that illustrated by the schoolroom example. Although air filtering and washing to remove dust and odors may make it practicable to reduce this ventilation to 2 or 3 air changes per hour, sanitary ventilation of crowded spaces calls for a very greatly increased air change, in rooms of high occupancy, utterly impractical by mechanical methods.

Sanitary Ventilation for High Occupancy

It is believed that for the prevention of the spread of respiratory diseases by airborne bacteria and viruses in places where there may be about 200-300 cu ft of space per person, an air purity such as that of summer living must be provided. It is believed that this could be approximated by ventilation at rates of 60 to 120 air changes per hour, or 1 to 2 changes per minute, if they were at all practicable from the standpoint of draft, noise and cost. Such ventilation rates would completely decontaminate a recently vacated room in 3 to 6 min, as shown graphically in Fig. 1 and would keep the concentration of bacteria in occupied and continuously contaminated rooms down to a safe level.

Sanitary ventilation can be added to ordinary ventilation by irradiating the upper air about 7 ft and sometimes the lower air (below 2 ft) with an intensity of 2537A ultraviolet adequate to provide zones of completely disinfected air from which a very slow general air circulation of 10-20 fpm, or 2-3 in. per second, provides the needed equivalent of 60-120 air changes per hour in the breathing zone. This method of sanitary ventilation by upper air irradiation will be discussed in detail later in this paper.

LESS SANITARY VENTILATION FOR LOW OCCUPANCY

It is probable that the sanitary ventilation requirement varies about inversely as the cubic feet of space per occupant, which means that in places where there may be 2000 to 3000 cu ft of air per occupant, as in the home or in spacious offices, instead of 200-300 cu ft per person, as in standard school rooms, one-tenth as much ventilation may be needed and 6-12 air changes per hour may be sufficient and practicable although seldom supplied by ordinary ventilation. Such a limited sanitary ventilation can be supplied by air disinfection with ultraviolet sources enclosed in air ducts or in individual room type air circulators of adequate capacity.

ULTRAVIOLET AIR DISINFECTION IN DUCTS

Although it is easily possible to disinfect all the air handled by a duct to bacterial equivalence to outside air and thereby remove one of the objections to recirculated air, the mechanical limitations of practical duct capacities in the sanitary ventilation of rooms have been indicated. There still remain many cases where duct air disinfection is desirable, where the duct air is the principal source of room infection, where direct upper and lower air irradiation may be impracticable, as in low ceilinged buses and railway cars, or where it may be desirable to disinfect recirculated air for the purpose of obtaining increased sanitary ventilation, or to justifying decreased make-up air. Since the recirculated air is usually four or five times the volume of the make-up air, disinfection of the former will increase four or five fold any disinfecting value which may have been credited to the latter, or disinfection of recirculated air may leave little need for make-up air except for odor and carbon dioxide control. In hospitals duct air disinfection may make air conditioning and heating by recirculation practicable to an extent not now thought safe or permissible.

PRACTICAL DUCT CALCULATIONS

Because of the way in which air ducts may be used to define the exposure factors of *intensity* (distance from the source) and *time* (minutes or seconds of travel through the irradiated zone) they have been used to secure the fundamental data on air disinfection (2, 3, 4, 5, 6). From these data and by methods described elsewhere (7) Fig. 2 has been prepared to simplify calculations of the amount of 2537A ultraviolet required for various air speeds and duct sizes, shapes and interior reflectivities. Because the absorption of 2537A energy by air, by bacteria or by any ordinary concentration of contamination is negligible, the output of a source is effective in producing lethal intensities through as many successive feet or in as many successive cubic feet as it can pass through before absorption by surrounding duct walls. This linear relationship between the average distance from the source to the duct walls and the cubic feet per minute (cfm) of air per watt of 2537A energy is fundamental to Fig. 2 and the effect of duct wall reflectivity and the increase of one of the duct dimensions is to increase the effective average distance the energy can travel before absorption and to increase the bactericidal effectiveness of the energy from the source. In laboratory work with one or two sources in

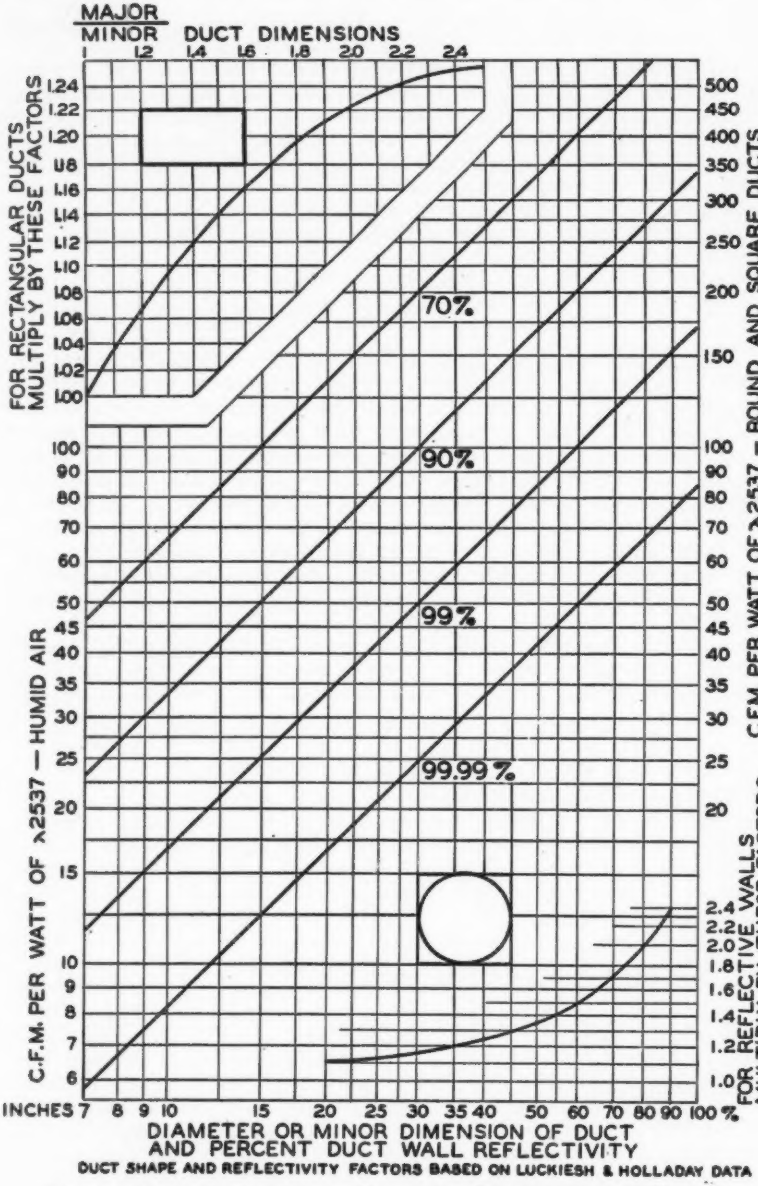


FIG. 2. CHART FOR CALCULATING ULTRAVIOLET REQUIREMENTS FOR VARIOUS PERCENTAGES OF KILL IN AIR DUCTS

large diameter ducts non-uniformity of ultraviolet intensity through the cross section of the duct must be taken into account, but in practical work the number of sources used provides sufficient uniformity of irradiation and permits a choice of various ways of placing the sources in the ducts. In the practical use of Fig. 2 the cfm per ultraviolet watt of energy, based on the minor dimension of a duct, must be increased by a multiplying factor indicated at the upper left for a rectangular duct and by another factor indicated at the lower right for reflective duct walls. For example, in a 30 x 60 in. duct with 60 per cent reflective walls and a rating of 12,500 cfm, a 99 per cent kill of bacteria would be produced in 50 x 1.22 x 1.5 or 91.5 cfm per watt of 2537A ultraviolet from the sources. This figure divided into 12,500 indicates a total of 137 ultraviolet watts which could be supplied by about 20 of the largest available germicidal lamps. This is provision for the most adverse conditions and it should be noted that the lethal exposures assumed by Luckiesh and Holladay (5) would indicate the use of but one half as much ultraviolet for the desired kill and lethal exposures assumed by Wells (10) would indicate the need for still less ultraviolet for the low relative humidity characteristic of the winter respiratory disease season. The writer suggests the use of the Luckiesh-Holladay value, or cfm ratings double those of Fig. 2, for practical work under average conditions.

Room Air Disinfection

Since any room or enclosed space ventilated by a recirculating duct system may be thought of as an extreme enlargement of the duct itself, the advantage of large duct dimensions suggests placing ultraviolet sources directly in rooms needing sanitary ventilation. Not all of this advantage can be realized, however, because ultraviolet intensities adequate to produce directly a sufficiently rapid bacterial disposal are also irritating to the eyes during the usual period of room occupancy and must be confined to the space above about 7 ft, with the addition in some cases of a space below about 2 ft (8). Such a method may be thought of as a 2 stage process in which intensive irradiation above a 7 ft level provides a reservoir of germ-free air which, in turn, serves to disinfect or dilute the contamination of the lower air as if by ordinary ventilating circulation or interchange of air. The usual first impression is that the interchange between these upper and lower zones is not sufficiently rapid to provide an effect of 60-100 air changes per hour in the lower zone. This impression often arises from a knowledge of the low rate of room air change provided by small ducts circulating air at a speed of 1000 fpm. Fig. 3 shows how effective low speed vertical air movement through one half the cross section of a room at various levels may be, assuming a provision of ultraviolet sufficient to maintain a constant upper air disinfection. Such air speeds are very difficult to measure but may be observed locally in the movement of smoke in a light beam. This effective air interchange results from a great variety of components even under the simplest conditions. An important addition to such measureable circulation as may be provided by a ventilating or heating system, is the air movement caused by window and door leakage, by cold walls and windows, by heated radiators and light sources, by the body heat and breath of all occupants and by the movements of occupants within the room.

Special attention is directed to the fact that most of this convective circu-

lation is essentially vertical in direction and much of it is increased by the temperature and weather conditions of the respiratory disease season. The amount of this air interchange parallels to some extent the need for it. On the other hand, under theoretical conditions of no vertical or horizontal air movement at all, there would be little general airborne transfer of infection from individuals to the group even though there still remained the droplet

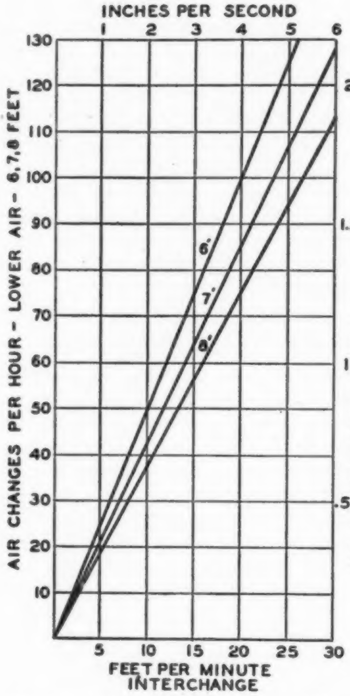


FIG. 3. THE RATE OF SANITARY VENTILATION IN LOWER AIR OF ROOM BY INTERCHANGE WITH COMPLETELY DISINFECTED UPPER AIR

transfer problem (sometimes thought solved by the face mask) and some very slow transfer by diffusion.

Wells (9) has published some data and he and others of the listed references have much unpublished data on bacterial concentrations in upper and lower air zones, which can in time be interpreted in terms of air interchange between the zones. Much additional data must be taken to cover all the practical combinations of spaces, ventilation, seasonal temperatures and occupancy. There

are bacterial and clinical data to indicate that the air movement in school and hospital rooms, calculated from air samples, is adequate and is equal to that indicated in Fig. 3. Further experience, however, may indicate the desirability of some additional air movement, such as could be produced with center, ceiling hung, large diameter, low speed air circulators operated concurrently with the ultraviolet sources.

UPPER AIR IRRADIATION

Wells (10) has analyzed the problem of upper air irradiation in detail by integrating through typical spaces the products of distances and intensities to secure a measure of total effectiveness. Assuming an energy in watts per square foot which will kill 63.2 per cent of the bacteria in 1 min as a unit lethal intensity and this amount of kill as a unit of kill, the Lethe, he has emphasized *ray length* and has developed a concept of watts/ft^2 per foot of distance, or Foot Lethes of irradiation, producing *Lethes* of bactericidal killing equivalent to air changes by ventilation. Lethes per minute or per hour of air disinfection, or sanitary ventilation, become equivalent to changes per minute or hour of air dilution by ordinary ventilation. As already noted, he has interpreted data, in terms of air circulation, on the differences in bacterial concentration in upper and lower air zones, with and without upper air irradiation, working with B-coli atomized into the air to permit an accuracy in air sampling difficult to secure when working only with naturally occurring bacterial contamination. Luckiesh and Holladay (11) have extended their accurate measurements in ducts to the room condition by considering it as divided by a partial horizontal diaphragm, in the plane of the ultraviolet sources, into upper and lower ducts in series, with air speeds through them inverse as their cross section, an air movement which they point out must be provided by artificial means if not naturally adequate.

In view of the basic importance of this upper and lower air interchange the writer tentatively offers a slight modification of the duct analogy by treating a whole room as a large vertical duct in which there is a continuous vertical interchange of air through the horizontal plane of the ultraviolet sources. This interchange may be represented as an upward movement through one half of the horizontal cross section of the room and a corresponding downward movement through the other half, considered as: *a.* a duct whose minor dimension determines the basic spatial effectiveness of the ultraviolet as in an ordinary duct but whose major dimension does not practically modify this rating because of the transverse placement of the sources and their essentially elliptical iso-intensity characteristics in the plane of the tube shown in Fig. 9; *b.* a duct in which the wall reflectivity must be kept below 10 per cent and otherwise may be disregarded; *c.* a duct whose effective length may vary over a wide range determined by the ceiling height of the room and which must be taken into account in all calculations; and *d.* a duct in which the air movement is widely variable (for reasons already discussed) and which must be assumed, for purposes of calculation, from such empirical data as are available. Fig. 4 extends the duct data of Fig. 2 to the larger cross sections and the lower air speeds characteristic of such room conditions and may be used for practical work in a similar way. Cfm ratings per watt of 2537A energy are modified by two multiplying factors covering the per cent output

efficiency of the germicidal fixtures used and their per cent effectiveness as determined by the inter-relationship of their distribution characteristics and various ceiling heights. These factors are both inherently characteristic of the various individual fixtures and should eventually be made available by the manufacturers through recognized testing laboratories. The only other variable unknown essential for practical work with Fig. 4 is the air interchange

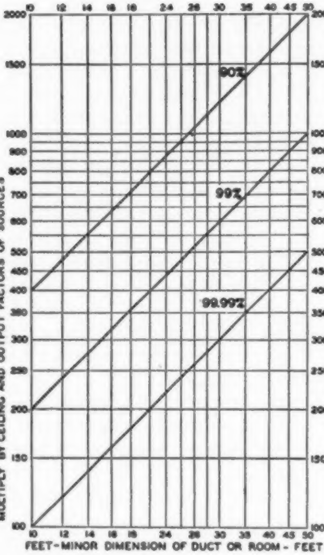


FIG. 4. CHART FOR CALCULATING ULTRAVIOLET REQUIREMENTS FOR VARIOUS PERCENTAGES OF UPPER AIR ROOM DISINFECTION

factor which is characteristic of each installation and is dependent upon a great variety of local conditions already mentioned.

AIR INTERCHANGE FACTORS

Because the actual air movement is random over the whole room area and in every possible direction it is impracticable to attempt direct measurement of air interchange factors or effective speeds of air movement vertically through one half the room cross section. These factors can only be determined indirectly by calculation from such measurements as Wells (9) has made of the lower air dilution resulting from the interchange with upper air disinfected to various measurable degrees. Although considerable air sam-

pling data subject to such an interpretation have been taken little data have been published. Until much more data covering a full range of ventilating and climatic conditions are available tentative factors only can be used and the following are suggested: schools 15-30, hospitals 15-25, offices and factories 10-15. The direct dependence of lower air disinfection by dilution upon the air interchange factor, shown in Fig. 3, was that which is characteristic of various design conditions in which the amount of ultraviolet energy is varied

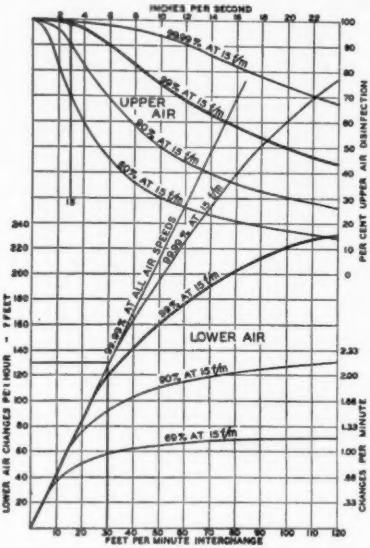


Fig. 5. Effect of Varying Air Interchange on Upper Air Disinfection and Lower 7 ft Air Sanitary Ventilation

in proportion to the air movement to maintain a uniform per cent of upper air disinfection. In practice, the amount of upper air disinfection is defined in the design of any one installation by the assumed average air interchange factor from which the cubic feet of air per minute in circulation is calculated and by the amount and effectiveness of the ultraviolet energy provided in accordance with Fig. 4. It is, therefore, important to know how the variations in the air interchange factor (produced by seasonal temperatures, movement of occupants, fans, etc.) affect the disinfection of the upper air and also how they affect its effectiveness, in turn, as a diluent of the lower air. This complicated interrelationship is shown graphically in Fig. 5 which represents a condition of equilibrium such that the destruction of bacteria above equals their introduction below and in the combined process there are surviving bacteria only because of the time interval between their introduction below and their exposure to ultraviolet energy above and because there is also a time lag

inherent in their coincidental logarithmic death rate above and the dilution rate below. While the per cent of upper air disinfection represented in the upper part of Fig. 5 can be calculated from the available duct data, only the rate of decontamination below can be predicted. The actual reduction in bacterial concentration below can be calculated only from a knowledge of the rate

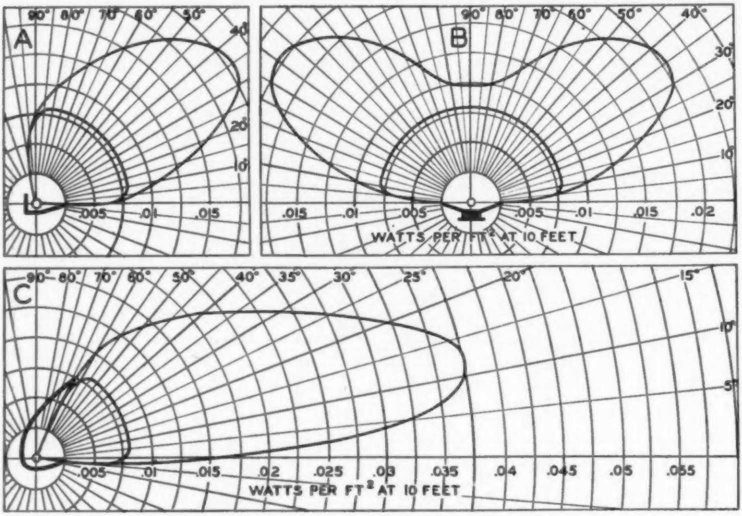


FIG. 6. SPATIAL DISTRIBUTION CURVES IN VERTICAL PLANE OF TYPICAL BACTERICIDAL LAMP FIXTURES

of contamination characteristic of each individual local condition and reliable data on this are only to be had by direct air sampling methods.

It is to be noted that an increase of air movement decreases the per cent of disinfection of the upper air by irradiation and, proportionately, its effectiveness as a diluent but, in spite of this, the increased air movement increases the rate of lower air disinfection by dilution. Fig. 5 also shows that, although sanitary ventilation is proportional almost solely to the air movement at lower rates, it becomes, at the higher rates of air movement characteristic of winter temperatures and the respiratory disease season, very dependent upon and is limited by the degree of upper air disinfection and by the amount and effectiveness of the ultraviolet irradiation.

PER CENT OUTPUT, FIXTURE EFFICIENCY

Fig. 6 presents the distribution curves in conventional form in a plane perpendicular to the tube of 3 typical bactericidal lamp fixtures equipped with

TABLE 1—OUTPUT AND EFFECTIVE OUTPUT FACTORS FOR 3 TYPICAL BACTERICIDAL LAMP FIXTURES WITH REFLECTORS SHOWN IN FIG. 6

	A	B	C
Output of bare germicidal lamp per cent	100	100	100
Output directly from lamp (Fig. 6) per cent	27	50	20
Output from reflector only (Fig. 6) per cent	18	35	32
Output total effective from fixture (Fig. 6) per cent	45	85	52
Effective Output Factor (Fig. 4)	0.90	0.85	1.04

aluminum reflectors. The figures for watts per square foot at 10 ft may be converted to microwatts/cm² at 1 meter by multiplying by 10,000. The inner curve represents the energy obtained directly from the tube only, while the outer curve represents the total from the tube and reflector. Distribution curves in the plane of the tube are all similar in shape and approximately circular and therefore are not shown. For practical purposes an index of total output may be secured by adding the intensity ratings at 5 deg intervals and dividing the sum by the product of the bare tube rating by the number of 5 deg intervals around it. Table 1 was calculated in this manner from the original curves of Fig. 6. The low effective output of Type A, in comparison with B, is due to energy lost to the increased reflector surface and to the side-walls above the unit, losses which are greater than the total loss to the extensive reflecting surface of Type C.

EFFECTIVE OUTPUT FACTORS

The effective output factors of fixtures, such as Type C, Fig. 6, with an exit angle of 90 deg or less, is double the per cent output because the effectiveness of such fixtures is not modified by the nonreflecting wall on which they may be mounted. This is in contrast with the bare sources, on which Fig. 4 is based, one half of whose output is absorbed when they are mounted in contact with a nonreflecting wall. For a bare lamp or a fixture such as Type B, Fig. 6, having an output distribution curve which is symmetrical relative to the duct or room axis and walls, the per cent output is also the *Effective Output Factor*. The effectiveness of such units, the integrated product of output and distances in all directions, remains practically constant regardless of their location from the center to contact with the side walls because the decreasing distances to the near walls are compensated by the increasing distances to the far walls.

TABLE 2—CEILING FACTORS

	FEET—DISTANCE FROM FIXTURE (LAMP CENTER) TO CEILING														
TYPE	1	2	3	4	5	6	7	8	9	10	11	12	13	18	23
A	0.06	0.12	0.17	0.23	0.28	0.33	0.37	0.41	0.45	0.49	0.53	0.56	0.60	0.75	0.87
C	0.16	0.25	0.35	0.42	0.48	0.54	0.58	0.64	0.67	0.70	0.73	0.76	0.78	0.88	0.96

CEILING ABSORPTION FACTORS

Figs. 2 and 4 are based on the assumption that there is no ultraviolet absorption in the duct or room except that of the walls and with provision in Fig. 2 for reflective walls. In applying Fig. 4 to room conditions, however, ceiling absorption becomes important and equivalent to an absorbing medium through the cross section of a duct. To illustrate this and to facilitate the calculation

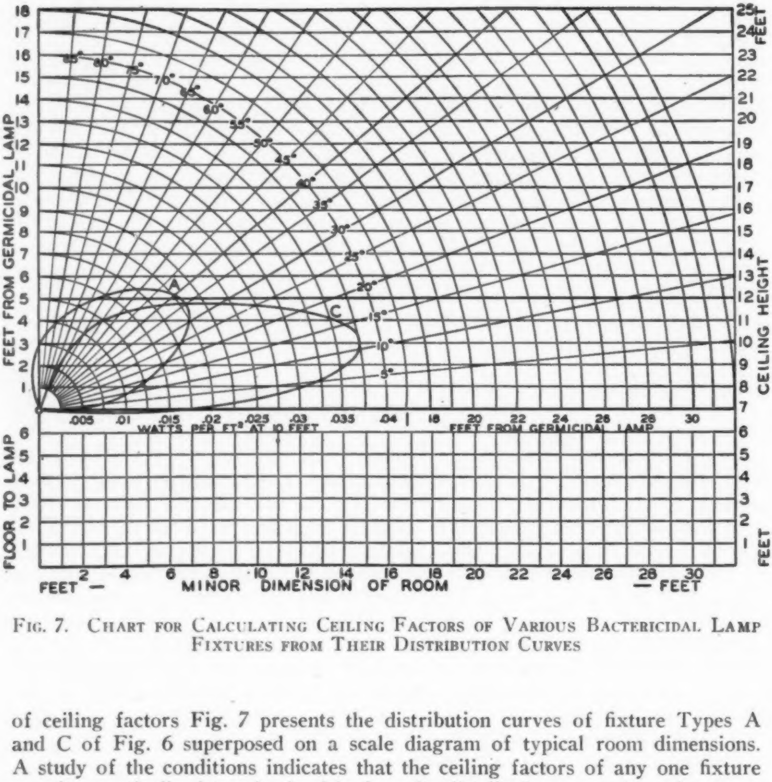


FIG. 7. CHART FOR CALCULATING CEILING FACTORS OF VARIOUS BACTERICIDAL LAMP FIXTURES FROM THEIR DISTRIBUTION CURVES

of ceiling factors Fig. 7 presents the distribution curves of fixture Types A and C of Fig. 6 superposed on a scale diagram of typical room dimensions. A study of the conditions indicates that the ceiling factors of any one fixture may be practically determined solely from its distribution curve in a plane perpendicular to the tube as was done in calculating the fixture output factor. Such ceiling factors are calculated as the ratios of the sums of the products of the intensities at 5 deg intervals and the corresponding distances to the side wall and ceilings at various heights, to the sum of the products of the intensities at 5 deg intervals and the corresponding distances to the side wall and to a theoretical ceiling at the side wall distance. Table 2 is for ceiling absorption factors so calculated from Fig. 7, on the basis of a 30-ft distance to the wall

and ceiling, and for the typical distribution curves of Fig. 6. The first three columns show the importance of even 6 in. differences in distance to the ceiling in the case of 8 and 9 ft ceiling heights where the lowest possible fixture mounting is indicated—even as low as 6 ft in some cases. The first 8 columns show the difference in effectiveness of the two types of fixtures under low ceilings, a considerable difference until the ceiling heights exceed 16 ft. These factors and the effective output factors should indicate to the most casual reader the importance of the germicidal fixture design in relation to its use. The use of louvers for appearance or use of inadequate reflectors may provide fixtures of practically no value, especially when used in rooms of ordinary ceiling height.

DEPRECIATION AND LAMP VARIATION FACTORS

It is important to note that, because of the logarithmic nature of the ultraviolet bactericidal action, changes of source output due to depreciation and individual source variations usually expressed in per cent do not at all result in the same per cent changes in air disinfection as has been more fully discussed elsewhere (7). Thus, as an extreme example, a 50 per cent depreciation in source output would result in a decrease in air disinfection from 99.99 per cent to 99 per cent or from 99 per cent to 90 per cent, or from 90 per cent to 70 per cent, depending upon the initial per cent of kill. Too little is known, however, regarding the relative clinical values of the higher rates of kill represented by 99 per cent and 99.99 per cent, the latter representing, for example, a survival of 1 organism in 10,000.

UPPER AIR CALCULATION

Fig. 4 is based on bactericidal exposures effective under the more adverse conditions of practical use and may be used with but small additional factors of safety to cover source depreciation. For example, assume that a school room 22 ft by 32 ft with a 12 ft ceiling is to be equipped with Type C bactericidal fixtures mounted on the sidewalls 7 ft from the floor, or 5 ft from the ceiling. The output factor would be 1.04 from Table 1, the ceiling factor 0.48 from Table 2 and an air interchange of 15 fpm may be assumed for fall and spring conditions. The total effective air movement in the room is therefore 5300 cfm upward through one half the room cross section at the 7 ft level and downward through an equivalent area. For a room width of 22 ft Fig. 4 indicates a basic rating of 425 cfm per watt of 2537A energy from the ultraviolet sources used for a 99 per cent kill. This rating multiplied by 1.04 and by 0.48 becomes 212 cfm and this divided into 5300 indicates 25 watts of ultraviolet as if from bare sources, a requirement which could be amply met with 4 typical germicidal lamps rated at 7.2 watts of 2537A each, or with 9 smaller commercial units rated at 2.9 watts each. Reference to Tables 1 and 2 indicates that approximately double these numbers of fixtures of Type A would be required, the products of the output and ceiling factors being in the ratio of 0.50 to 0.25. This is provision for the most adverse conditions and it should be noted that the lethal exposures assumed by Luckiesh and Holladay (5) would indicate the use of but one half as much ultraviolet for the desired kill—and exposures assumed by Wells (10) would indicate still less ultraviolet

—for the low relative humidity characteristic of the winter respiratory disease season. The writer suggests the use of the Luckiesh and Holladay value or cfm ratings double those of Fig. 4 for practical work under average conditions.

LOWER AIR CALCULATION

Assuming such easily obtained and practically complete disinfection of the upper air by irradiation as is illustrated by the foregoing example, the corre-

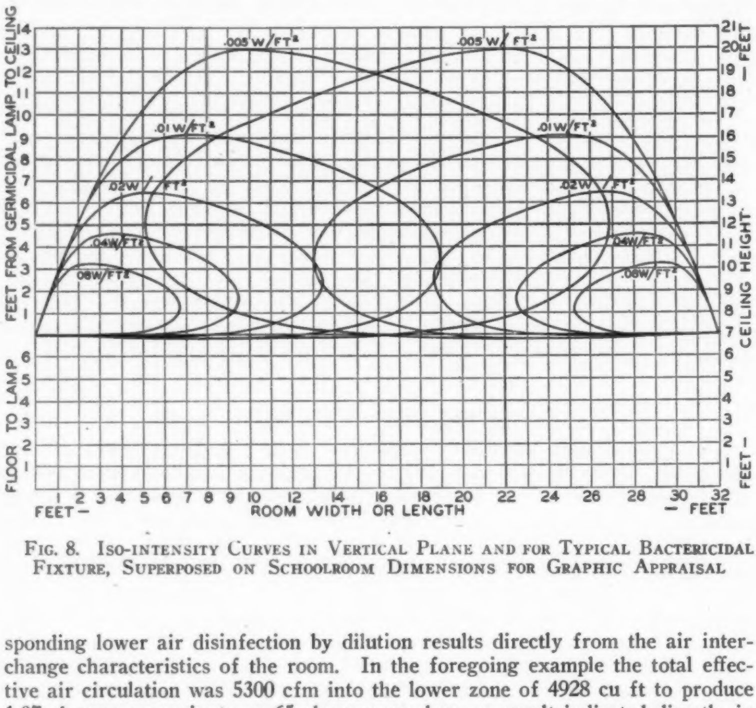


FIG. 8. ISO-INTENSITY CURVES IN VERTICAL PLANE AND FOR TYPICAL BACTERICIDAL FIXTURE, SUPERPOSED ON SCHOOLROOM DIMENSIONS FOR GRAPHIC APPRAISAL

sponding lower air disinfection by dilution results directly from the air interchange characteristics of the room. In the foregoing example the total effective air circulation was 5300 cfm into the lower zone of 4928 cu ft to produce 1.07 changes per minute or 65 changes per hour, a result indicated directly in Figs. 3 and 5 for a fixture mounting height of 7 ft and an air interchange rate of 15 fpm. The variation in disinfection with seasonal changes in this air interchange rate is shown in Fig. 5, where the above example is illustrated by the heavier lines and a design for a 99 per cent upper air disinfection at 15 fpm.

GRAPHIC FIXTURE APPRAISAL

Figs. 8 and 9 serve to illustrate the use of iso-intensity curves which are calculated for fixture Type C and are superposed on room dimensions as in a

typical standard school room installation for a graphic appraisal of the distribution of ultraviolet in space. Note that the curves of Fig. 8 are in the plane of maximum intensity 10 deg-13 deg from the horizontal and that in both cases energy incident at any given point in space from different directions is fully additive in a sense not true in illumination practice although such sums are

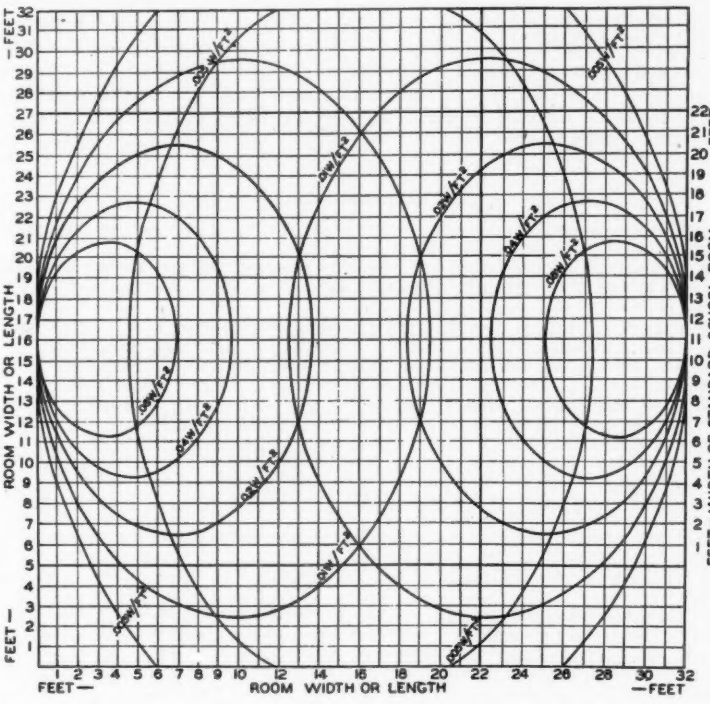


Fig. 9. Iso-intensity Curves of Typical Bactericidal Fixture in Plane of Tube and Direction of Maximum Intensity, Superposed on Typical School-room Dimensions for Graphic Appraisal

not indicated in Figs. 8 and 9. The relationships between various ceiling heights and the iso-intensity curves should be noted especially in Fig. 8.

Epidemiological and Clinical Factors

When sufficiently sensitive methods of air sampling are available to permit determination of actual concentrations of pathogenic bacteria and viruses in occupied rooms and when such data are correlated with epidemiological and clinical data on the incidence of respiratory disease the data on per cent kill

in this paper will indicate how to secure and maintain a space concentration below any specified critical value. The writer believes a specification of such critical values and the interpretation or discussion of sanitary ventilation from the standpoint of room occupancy, occupant infection and susceptibility and the virulence of the current respiratory disease to be a part of the bacteriological and epidemiological basis rather than the physical basis of air disinfection.

Although the physical factors covered in this paper are fundamental to any proper use of ultraviolet air disinfection their final application must be further conditioned on equally important factors related to the prevalence of respiratory disease and the susceptibility and crowding of the people involved. To whatever extent these bacteriological and epidemiological factors are uncertain assumption of the worst probable conditions is suggested in the provision of correction by sanitary ventilation.

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A NEW FRICTION CHART FOR ROUND DUCTS

By D. K. Wright, Jr.,* Cleveland, Ohio

This paper is the result of research carried on by the American Society of Heating and Ventilating Engineers at its Research Laboratory, Cleveland, Ohio

INTRODUCTION

This paper is the result of preliminary studies made by the Society's Research Laboratory. The work was carried out under the general direction of the Technical Advisory Committee on Air Distribution and Air Friction † and the specific direction of a subcommittee consisting of R. D. Madison, chairman; S. H. Downs, L. P. Saunders, K. H. Flint, H. F. Hagen and L. L. Simmons.

Purpose of This Report

One of the important problems attending the design of any ventilating or air conditioning system is that of providing satisfactory duct work. In some instances this problem means the balancing of first costs and operating costs; in others it means obtaining the required flow of air with low enough fan and air speeds to avoid objectionable noise. A vital part of the solution rests in a satisfactory estimate of the flow resistance offered by the system.

The total resistance to the flow of air through an air handling system is made up of the following: 1. resistance of straight duct runs; 2. resistance of fittings and transformations; 3. resistance of grilles, registers and dampers for controlling air flow; 4. resistance of equipment—coils, filters, etc.

This paper is concerned only with the resistance of straight ducts and is further restricted to round ducts. It is the result of a survey and subsequent discussion of the problem by the Subcommittee on Air Duct Friction of the Society's Technical Advisory Committee on Air Distribution and Air Friction. The important companion subject of conversion from round to rectangular cross-sectional duct of the same capacity and resistance is to be investigated and reported on later.

In 1938 a research project was initiated at the A.S.H.V.E. Research Laboratory (then in Pittsburgh) to provide basic data on the resistance to flow of air in sheet metal ducts, for the purpose of checking the friction chart then in general use by the Society. The Laboratory made an extensive series of tests on round ducts, 4, 8 and 24 in. in diameter, with and without joints and on several sizes of rectangular ducts. Two papers were presented, one in

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1939 [1] and the other in 1940.[2] The former contained two friction charts developed from the experimental data and in 1942 one of these charts (that is, the one for straight round duct with 40 joints per 100 ft and with no factor of safety allowed) was incorporated in THE GUIDE and has been used since that time.

From the time of publication of the first paper, it was known that the Laboratory values were somewhat lower than duct friction values in general use and, in the case of the ducts without joints, lower than any previously reported. Criticism of the chart indicated the advisability of a resurvey of the problem, which was undertaken by the Committee on Air Distribution and Air Friction, resulting in the present paper. Its purpose is to offer a new air friction chart with an explanation of the basis on which it has been constructed.

BASIC THEORY OF FLOW IN CLOSED PIPES

The problem of resistance to flow of a fluid in a closed straight round conduit is most simply attacked by the use of non-dimensional groupings of variables as follows:

$$\text{If } h = f\frac{L}{D}\frac{V^2}{2g} \quad \ldots\ldots \quad (1)$$

where

h = head loss due to friction in feet of the fluid flowing.
L = conduit length in feet.
D = conduit inside diameter in feet.
V = fluid velocity in feet per second.
g = acceleration due to gravity, 32.2 fps squared,

then f is the non-dimensional friction coefficient which for isothermal flow at the velocities used in ventilating work is known to depend on just two other non-dimensional groups; the Reynolds Number, $R = \frac{VD}{\nu}$ (ν = kinematic viscosity of the fluid in sq ft per second) and the relative roughness, $\frac{\epsilon}{D}$ of the conduit surface, where ϵ is a roughness measure having the same dimension as that used for the diameter, D.

For very low Reynolds Numbers—below 2000—the fluid motion is laminar and f is found to depend only on R being independent of roughness as long as it is not so extreme as to cause an effective reduction in diameter. This range is relatively unimportant in ventilating work. For an intermediate range of Reynolds Numbers, f depends on both R and

$$\frac{\epsilon}{D}$$

while for high Reynolds Numbers, the flow becomes completely turbulent, f becomes independent of R and depends only on relative roughness. Almost all the operating conditions under discussion here fall within the intermediate

[1] Exponent numerals refer to Bibliography.

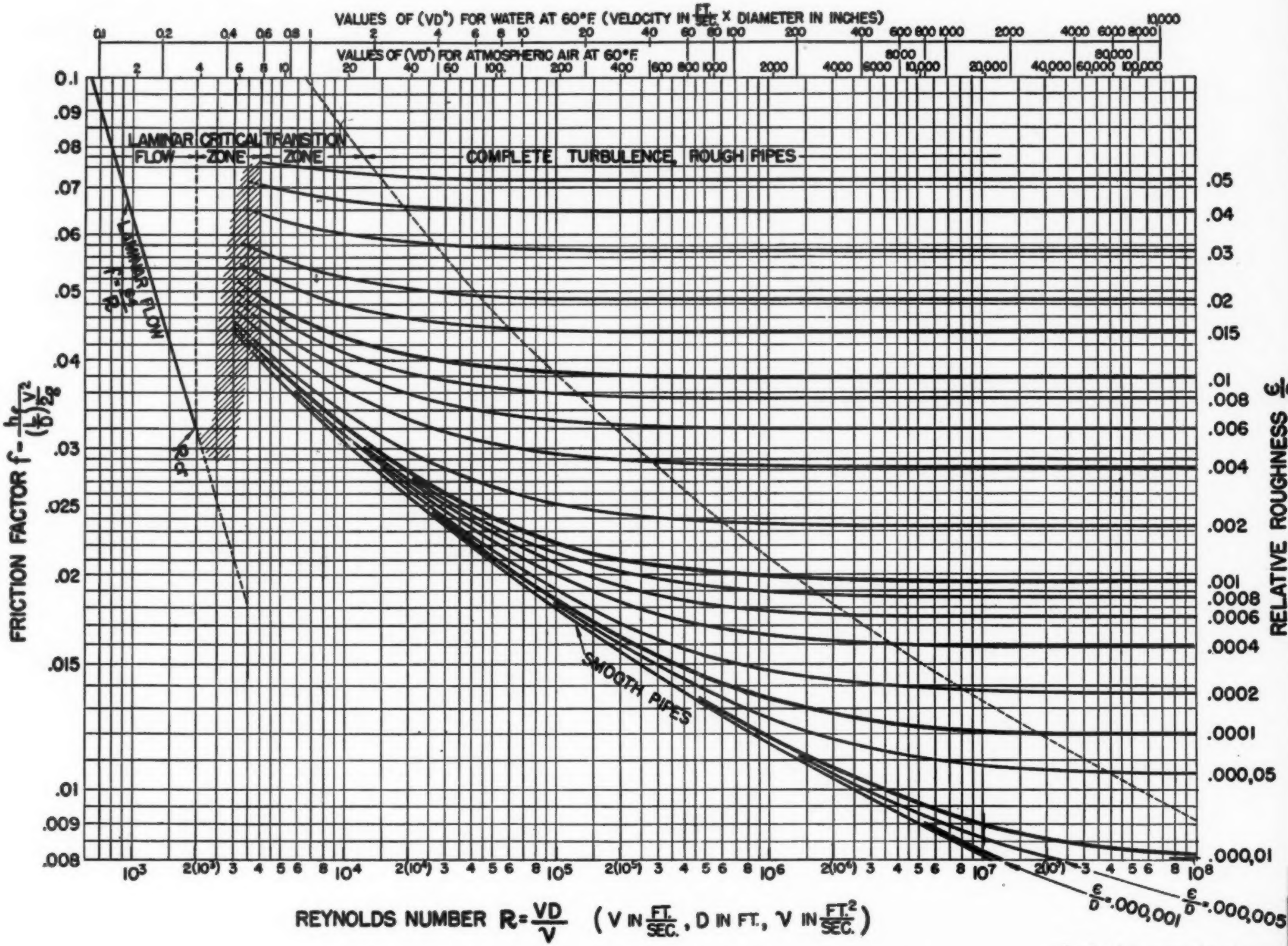

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Fig. 1. Friction Factor in Relation to Reynolds Number and Relative Roughness

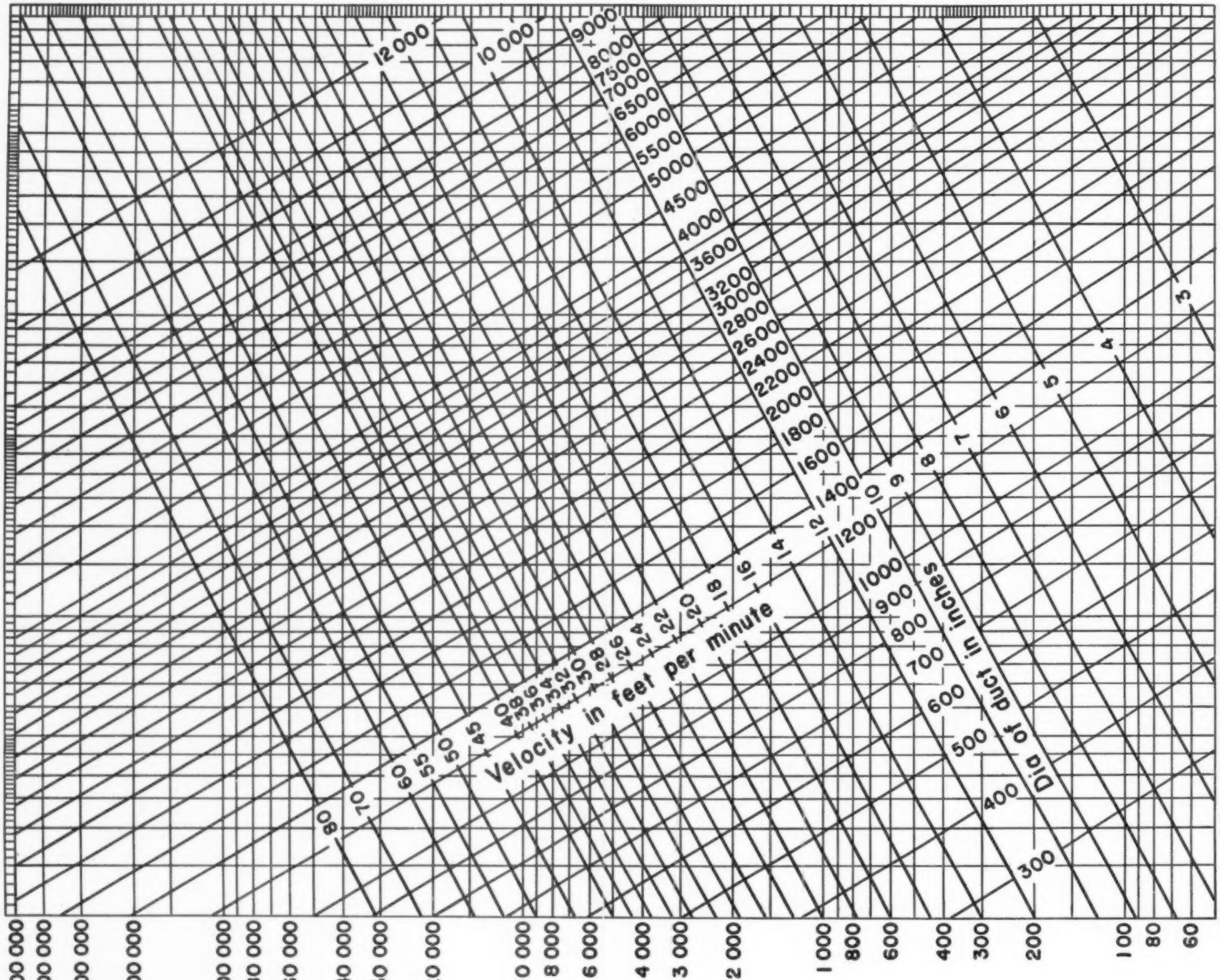

FIG. 2. AIR FRICTION CHART FOR ROUND STRAIGHT GALVANIZED DUCT OF AVERAGE CONSTRUCTION WITH ABOUT 40 JOINTS PER 100 FT
BASED ON DRY AIR AT 29.92 IN. BAROMETER AND 70 F

transition region between laminar and completely turbulent flow. The variations of the friction factor with both Reynolds Number and relative roughness are, therefore, of importance in the present discussion.

One of the early investigators to recognize the dependence of the friction coefficient on Reynolds Number and to use the now familiar f versus R chart was Blasius [3] who, in 1913, plotted a considerable body of experimental data for smooth drawn pipes at values of R between 3000 and 100,000 on log-log coordinates. He decided that a straight line fitted the data and proposed an equation of the type,

$$f = \frac{A}{R^a} \quad \ldots \quad (2)$$

This relation is the basis for the commonly seen equations giving the head loss in terms of the velocity of flow to some power less than 2, for if it is substituted in Equation (1), we obtain

$$h = \frac{A}{R^a}\frac{L}{D}\frac{V^2}{2g} = \frac{A\nu^a}{(VD)^a}\frac{L}{D}\frac{V^2}{2g}$$

$$= KL\frac{V^{2-a}}{D^{1+a}} \quad \ldots \quad (3)$$

Values of a in common use run from 0.14 to 0.20.

The effect of change of relative roughness with duct diameter is frequently accounted for by further modifications of the exponent of the diameter. This equation is the basis for most of the air duct friction charts now in existence. Its use amounts to an assumption that the log-log f-R chart is made up of a series of parallel straight lines, one for each relative roughness.

In contrast, *test results* have indicated that the f-R chart should not be such a series of straight lines, but rather a set of curves, concave upward, fanning out and approaching zero slope at very high Reynolds Numbers. A short length of any one of these curves may successfully be approximated by a straight line as found by Blasius, but when the range of Reynolds Numbers and relative roughnesses covered by the usual friction chart is considered, it is not hard to understand the difficulties encountered with a chart based on Equation (3).

In 1933, the results of the analysis by Pigott [4] and Kemler of most of the data available on pipe friction were published, the final form being a log-log f-R plot made up of various broken straight lines for different degrees of roughness. At about the same time, Nikuradse,[5] von Karman [6] and Prandtl [7] published theoretical and experimental work which defined the flow in smooth pipes and in rough pipes for complete turbulence. In 1939, Colebrook [8] succeeded in developing a functional form now believed to be satisfactory in the transition zone.

Taking advantage of these theoretical developments, Rouse [9] and, recently, Moody,[10] have rebuilt the Pigott chart. Moody's form, originally published in the *A.S.M.E. Transactions,* November, 1944, is reproduced here as Fig. 1. Since it is recognized by authorities in the field of fluid mechanics as being

based on the best information now available, this graph has been used in computing values for the new air friction chart.

BASIS FOR NEW CHART

In order to use Fig. 1, the Reynolds Number of the flow and the duct roughness must be known. The Reynolds Number has been computed for standard dry air at 29.92 in. Hg barometer and 70 F, taking the kinematic viscosity, ν, corresponding to these conditions at 16.3×10^{-5} sq ft per second.

To determine a satisfactory estimate of the roughness of average sheet metal duct, a survey of published test results on ventilating type ducts and of existing friction charts has been made. This investigation is summarized in Appendix I. As a result, the roughness measure, ϵ, of clean, round galvanized sheet metal duct with about 40 slip joints per 100 ft has been taken as 0.0005 ft and this value has been used in making up the chart as shown in the sample computation, Appendix II.

The proposed air duct friction chart is presented in Fig. 2. The conditions under which it may be used are stated in the accompanying caption. The effects of changes in air conditions or duct materials are discussed herewith.

TABLE 1—COMPARISON OF DUCT FRICTION CHARTS

AIR VELOCITY—FPM	SOURCE	FRICTION LOSS IN INCHES OF WATER PER 100 FT DUCT DIAMETER—IN.				
		5	10	20	40	80
500	1940 Guide	0.14	0.058	0.024	...	...
	1945 Guide[a]	0.11	0.045	0.019	...	...
	IHVE	0.11	0.044	0.018	...	...
	A	0.11	0.048	0.020	...	...
	B	0.093	0.047	0.023	0.012	...
	Fig. 2	0.11	0.044	0.018	0.0080	...
1,000	1940 Guide	0.51	0.21	0.086	0.036	...
	1945 Guide[a]	0.40	0.17	0.066	0.027	0.011
	IHVE	0.38	0.16	0.066	0.028	0.011
	A	0.40	0.17	0.073	0.031	0.013
	B	0.38	0.19	0.095	0.050	0.023
	Fig. 2	0.37	0.16	0.068	0.029	0.012
3,000	1940 Guide	4.0	1.7	0.67	0.27	0.12
	1945 Guide[a]	3.0	1.3	0.51	0.21	0.085
	IHVE	...	...	0.49	0.21	0.082
	A	3.0	1.3	0.55	0.23	0.10
	B	3.4	1.7	0.85	0.42	0.21
	Fig. 2	3.0	1.3	0.54	0.24	0.10
10,000	1940 Guide	...	...	5.8	2.4	0.98
	1945 Guide[a]	...	11.	4.4	2.0	0.78
	IHVE	...	...	...	...	...
	A	...	12.	5.0	2.2	0.90
	B	...	...	9.3	4.6	2.3
	Fig. 2	...	13.	5.7	2.5	1.1

[a] Recommended 10 per cent safety factor included.

It will be noticed that the duct diameter and air velocity lines in Fig. 2 are curved. This is, of course, the result of using the curved friction factor *vs.*

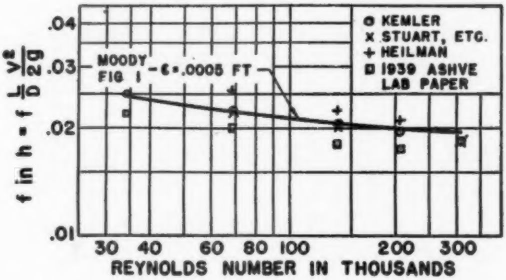

Fig. 3. Comparsion of Moody Data (Fig. 1) with Test Data—8 in. Galvanized Duct with Joints

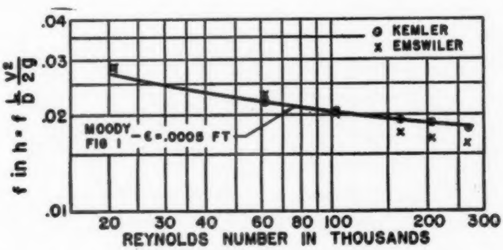

Fig. 4. Comparison of Moody Data (Fig. 1) with Test Data—12 in. Galvanized Duct with Joints

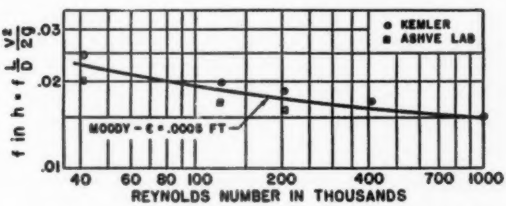

Fig. 5. Comparison of Moody Data (Fig. 1) with Test Data—24 in. Galvanized Duct with Joints

Reynolds Number relations from Fig. 1 rather than the straight line Equation (3). A comparison of the new chart with several others in use at the present time is given in Table 1 and in Figs. 6 and 7. The agreements and disagree-

ments are presented there more forcefully and more succinctly than can be done in words.

ADAPTATION OF CHART TO CHANGED CONDITIONS

Two types of changes might occur from those conditions assumed in constructing the chart; changes of duct material or roughness and changes of air

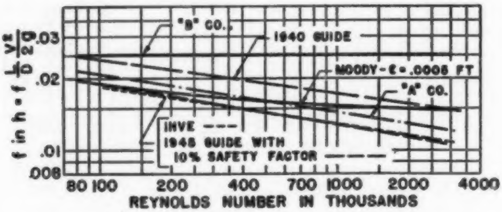

FIG. 6. COMPARISON OF MOODY DATA (FIG. 1) WITH FRICTION CHARTS—24 IN. GALVANIZED DUCT WITH JOINTS

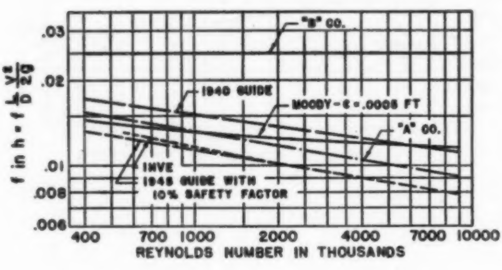

FIG. 7. COMPARISON OF MOODY DATA (FIG. 1) WITH FRICTION CHARTS—72 IN. GALVANIZED DUCT WITH JOINTS

conditions. Figs. 3, 4 and 5 give some indication of the deviations in duct roughness normally to be expected. The largest deviations of points from the curves drawn are of the order of 10 per cent. The ultimate in smoothness obtainable with sheet metal duct is of the order of $\epsilon = 0.000,005$ ft, a result obtained from some unpublished data by J. R. Weske on 6 in. galvanized duct with very carefully fitted flanged joints. This roughness is about the same as that for smooth drawn tubing. Friction is, therefore, very sensitive to joint frequency and construction.

Fig. 2 has been made up for average construction and might be expected to give results accurate to within ± 5 to 10 per cent. For duct materials having

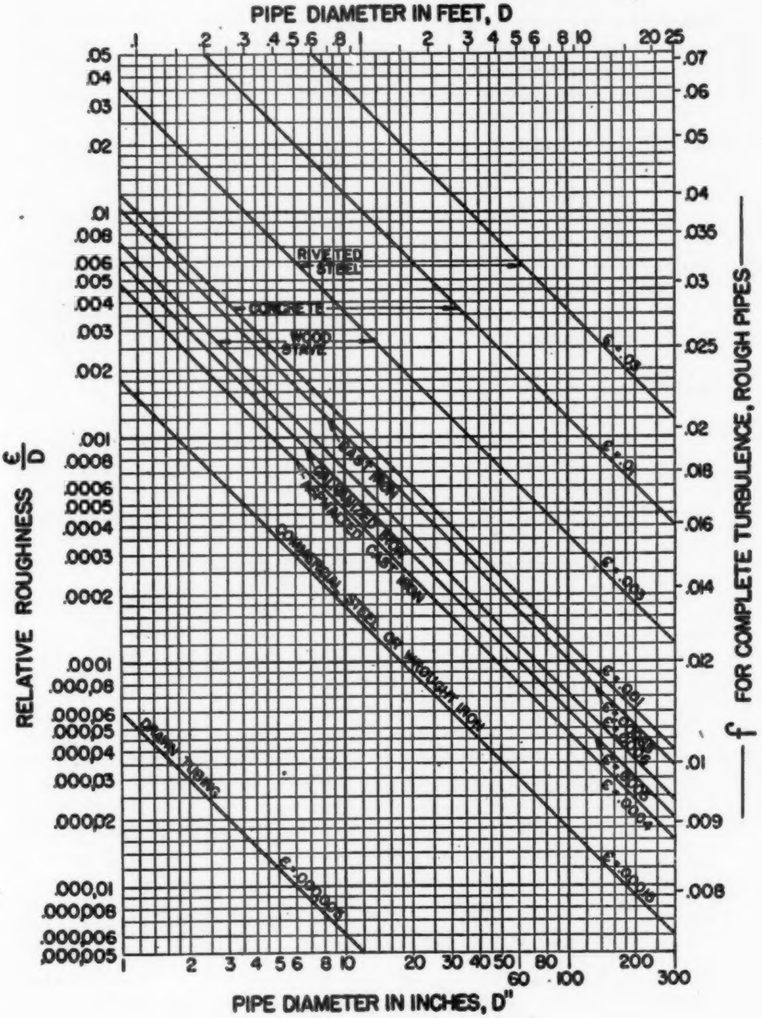

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Fig. 8. Roughness Factors for Various Pipe Materials.

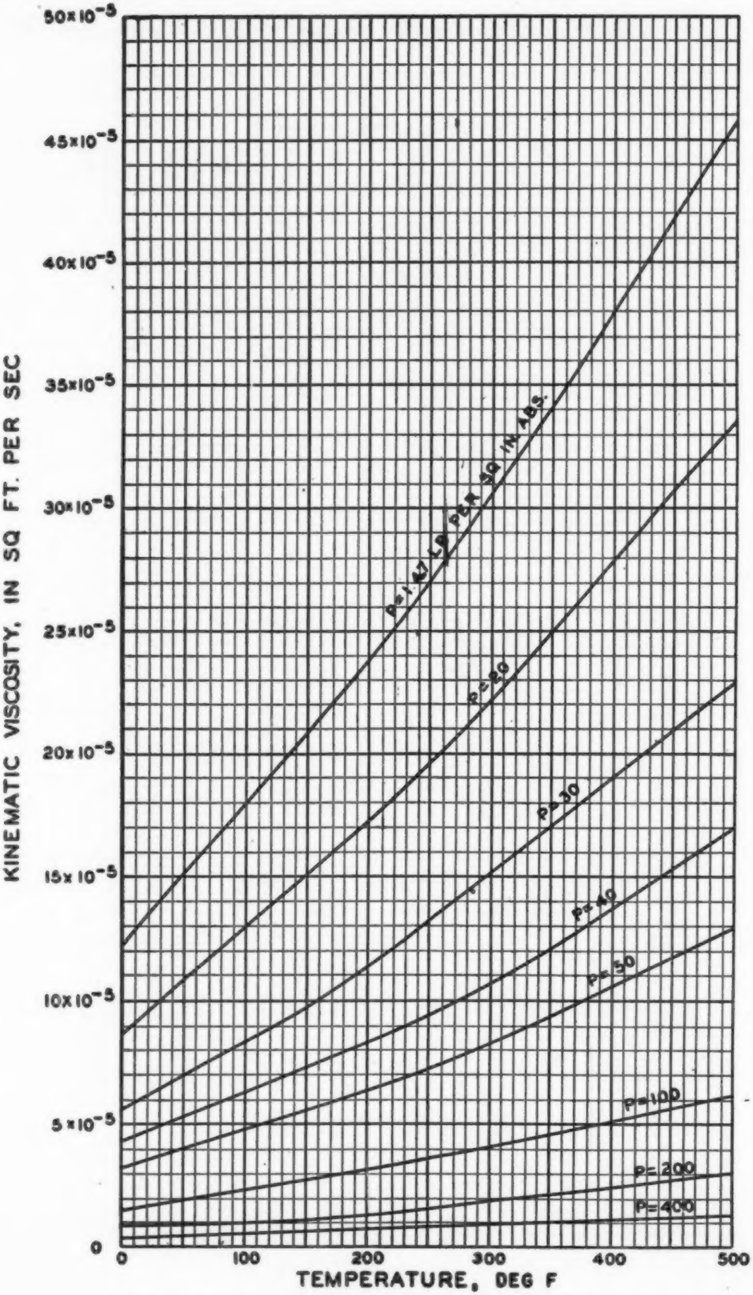

FIG. 9—CURVES FOR DETERMINING KINEMATIC VISCOSITY

greatly different roughness it has been common practice to apply a factor to friction loss values read from the chart; in other words, to make a constant percentage correction. Examination of Fig. 1 shows this procedure to be very undesirable since the change in friction factor from one line of relative roughness to another is small where the factors are high and is large where they are low. The best solution to the problem would be to have several different charts for the materials most frequently used or to work from Fig. 1 as indicated in Appendix II.

Changes in air conditions—temperature, pressure, humidity—affect friction values first, by changing the kinematic viscosity, hence, the Reynolds Number and the friction factor read from Fig. 1; second, by changing the specific weight used in the conversion of units from feet of air to inches of water.

Changes in humidity produce changes in kinematic viscosity and density which are small enough to be disregarded in duct design. Temperature and pressure variations affect kinematic viscosity and specific weight in opposite directions and thus tend to be compensated, although the change in specific weight is more powerful an influence on duct friction than the accompanying change in kinematic viscosity.

As a result, the following statements appear justified:

1. Usual variations in barometric pressure or in pressures required to circulate air through low pressure duct systems have negligible effect on duct friction.

2. Variations in air temperature of the order of ± 20 F from 70 F affect duct friction to a small enough extent to permit the use of Fig. 2 without correction.

3. The only theoretically correct method of making corrections for changes in either air conditions or duct roughness is recalculation from Fig. 1, using the new conditions. The Colebrook function, used in plotting Fig 1, is expressed as:

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{\epsilon/D}{3.7} + \frac{2.51}{R\sqrt{f}}\right)$$

a particularly cumbersome expression from the standpoint of separating the variables. It is unlikely that a simple means of correcting the duct friction chart can be worked out of this relation.

Summary

1. Following a survey of available literature and using the newest theoretical developments, a new air duct friction chart has been worked out and is presented in Fig. 2.

2. Because of a lack of test data on large ducts, the upper end of the chart is still subject to some uncertainty. The committee and the author are anxious to receive the comments and suggestions of those whose experience in this range may help to resolve the doubt.

3. No simple theoretically sound method exists for adapting the readings from the chart to extensive changes in air or duct surface conditions. A satisfactory approximate method may be possible, but remains to be developed.

Acknowledgments

The author gratefully acknowledges the help he has received from each of the following: The Technical Advisory Committee on Air Distribution and Air Friction, particularly the Subcommittee on Air Duct Friction, for discussion and suggestions on the decisions reached in the paper. Cyril Tasker and the staff of the Research Laboratory, in particular, Miss Doris Dietz who calculated and drew the friction

chart and prepared figures (except Figs. 1, 8 and 9) for publication. Prof. G. L. Tuve for helpful criticisms. Prof. L. F. Moody and the *American Society of Mechanical Engineers* for permission to reproduce Figs. 1 and 8. Prof. L. G. Miller and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS for permission to reproduce Fig. 9.

APPENDIX I

SUMMARY OF PUBLISHED DATA ON DUCT FRICTION

The purpose of this survey of the literature was to gather enough actual test data to decide what value of ϵ best represented the surface roughness of round sheet metal duct of average construction. An unsatisfactory small amount of such data was uncovered, none of it on ducts larger than 24 in. in diameter. This is naturally the result of the equipment size and expense involved in testing large sized ducts at high velocity.

It must be admitted that the decision reached is not as certain as might be desired, but it is felt that the final results are closer to the truth than those obtained in the past by simple extrapolation of the original data. The uncertainty lies in whether the absolute roughness, ϵ, of the duct surface is constant for all sizes or increases with increasing diameter. This is discussed in greater detail.

The data examined and presented here in Figs. 3, 4 and 5, were obtained from published work by Emswiler,[11] Kemler,[12] Heilman and McArthur,[13] Houghten, Schmieler, Zalovcik and Ivanovic[1] and Stuart, Warner and Roberts.[14] All except the material from Kemler are the results of actual tests by the authors. The values recommended by Kemler have been given equal weight because of his outstanding work with Pigott in correlating all the data available on pipe friction to 1933.[15] Prof. L. G. Miller has also used the Pigott chart for duct design, but his recommended friction coefficients are slightly higher than Kemler's.

In his original paper,[15] Kemler, reported some work by Petit on 10 and 13 in. sheet metal ducts which indicates much higher roughness ($\epsilon = 0.001$ to 0.005) than found by any of the others. Since Kemler has not given these results much weight, they have been disregarded.

The conclusion from the present survey was that a value of $\epsilon = 0.0005$ ft provided the best agreement with the data analyzed and the lines drawn on Figs. 3, 4 and 5 are taken from Fig. 1 to indicate that agreement.

Still another comparison, with air duct friction charts now in use, is given in Figs. 6 and 7. The method of comparison selected was to reduce the charts to their equivalent friction factors and to plot them with the appropriate curve from Fig. 1. The charts used were, that published in the A.S.H.V.E. GUIDE up to 1941; the new chart developed from Research Laboratory data in 1939 and used in THE GUIDE from 1942 to 1945; the chart of the *British Institution of Heating and Ventilating Engineers* and charts used by two companies in the field, labeled here, A and B. Since working values were obtained from all other charts, the recommended 10 per cent safety factors have been applied to those taken from THE GUIDE, 1945. The *B* chart is an old one computed from a constant friction factor of 0.025.

Figs. 6 and 7 have been drawn over the range of Reynolds Numbers covered by the charts. The effect of using a straight line, constant slope equation is clearly indicated here. In Fig. 6, the slopes of all the lines, and the magnitudes of THE GUIDE, 1945 and *I.H.V.E.* values are not greatly different from those of the curve of Fig. 1 at low Reynolds Numbers—the range in which test results are available. At the higher velocities, however, the deviation is serious and the results of too extreme extrapolation of test data are apparent.

Fig. 7 was drawn to compare the charts at the upper end where test results are lacking. The indication—that the values given in the chart in the 1939 paper (1942-

1945 GUIDE chart) are too low, that a better value would fall between A and THE GUIDE, 1940 charts and far below the B chart—is not unexpected and fails to throw any doubt on the proposed method.

CONCLUSIONS

1. For duct sizes up to 24 in., test results indicate that round, galvanized, sheet metal duct with approximately 40 joints per 100 ft has a surface roughness represented by $\epsilon = 0.0005$ ft on Fig. 1.

2. No test results are available at large diameters to indicate whether surface roughness changes with diameter or not. Comparison with existing friction charts with consideration of the reputations of those charts furnishes a rough check that the assumption of constant ϵ may not be far from the truth.

APPENDIX II

SAMPLE CALCULATION

As an example of how the new chart, Fig. 2, was computed and to indicate the method of using the basic data presented here, the following problem will be solved:

What is the friction loss in inches of water of dry air at 70 F and 14.7 psi absolute flowing through average clean sheet metal duct 100 ft long and 16 in. in diameter at a velocity of 1400 fpm?

The duct diameter in feet is $16/12 = 1.33$.

For average sheet metal duct assume $\epsilon = 0.0005$ ft. Then the relative roughness,

$$\frac{\epsilon}{D} = 0.0005/1.33 = 0.000375$$

(This value can also be obtained from Fig. 8 which is reprinted from Moody's paper in the *Transactions of the American Society of Mechanical Engineers.*[10] Fig. 8 also suggests values of ϵ to be used for a number of other pipe or duct surfaces.)

The Reynolds Number, R, of the flow must be determined in order to read the friction factor from Fig. 1.

$$R = \frac{VD}{\nu}$$

where

$$V = \text{fps} = \frac{1400}{60} = 23.3$$

$$D = \text{ft} = 1.33$$

$$\nu = \text{Kinematic viscosity, sq ft/sec.}$$

The kinematic viscosity can be read from Fig. 9, reprinted from L. G. Miller's paper on duct design[16].

For 14.7 psi absolute, and 70 F,

$$\nu = 16.3 \times 10^{-5} \text{ (from Fig. 9)}$$

Hence

$$R = \frac{VD}{\nu} \times \frac{23.3 \times 1.33}{16.3 \times 10^{-5}} = 1.91 \times 10^{5}$$

From Fig. 1 corresponding to

$$R = 1.91 \times 10^{-5} \text{ and } \frac{\epsilon}{D} = 0.000375$$

read $f = 0.0182$

Then the head loss in feet of air,

$$h = f\frac{L}{D}\frac{V^2}{2g} = 0.0182\,\frac{100}{1.33}\,\frac{(23.3)^2}{2 \times 32.2} = 11.6$$

The specific weight of air may be computed with sufficient accuracy from the perfect gas equation:

$$\frac{W}{V} = \frac{P}{RT} = \frac{14.7 \times 144}{53.3 \times 530} = 0.075 \text{ lb per cu ft.}$$

If the specific weight of water may be taken at 62.3 lb per cu ft, the friction loss in inches of water,

$$H = 11.6 \times \frac{0.075}{62.3} \times 12 = 0.167$$

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DISCUSSION

R. D. MADISON, Buffalo, N. Y. (WRITTEN): Professor Wright has presented the Society with a new friction chart which I am sure we can use with confidence as representing the latest. The chart has the same general appearance and for that reason should be familiar to most users of THE GUIDE. However, the lines are rotated in position and curved to fit the Moody-Rouse data described by the author and therefore the data are brought into closer agreement with rational and acceptable values.

As the chart is made up for round ducts, the point is raised as to how to use it for other conditions, such as conduits of different roughness as well as for shapes other than round. Theoretically the formula now in THE GUIDE for round and rectangular equivalents should be changed to take care of these new data. Since it is based upon the exponential law, it would be necessary to express the data of the friction chart by an exponential equation that most nearly approximates true data. The following formula does this over the body of the chart with an error of 4 or 5 per cent.

$$\Delta p = 2.74 \frac{L}{d^{1.22}} \left(\frac{V}{1000}\right)^{1.90}$$

where

Δp = pressure loss in inches of water.
L = length of duct in feet.
d = diameter of duct in inches.
V = velocity of standard air in feet per minute.

If we assume that the loss in a rectangular duct is:

$$\Delta p_1 = 2.74\, L \left(\frac{a+b}{2ab}\right)^{1.22} \left(\frac{V_1}{1000}\right)^{1.90}$$

on the basis of the mean hydraulic radius, then equating the friction in two ducts having equal volumes of flow, we have:

$$d = \frac{1.297(ab)^{0.621}}{(a+b)^{0.242}}$$

Rectangular equivalents of round ducts as given by this formula compared with that given in THE GUIDE are very nearly the same for ducts that are square or nearly so. There begins to be a difference where the ratio of a/b is large. However, since the validity of the formula for mean hydraulic radius is under some question for large aspect ratios, there seems to be no need of recalculating THE GUIDE tables until a study is made of this question. The Committee on Duct Friction has this in mind as one of the projects for the Laboratory.

The question of roughness of surface is too complicated to be represented by simple factors. However, a plot can be made of factors versus diameter for various velocities to obtain a family of curves that are fairly well separated and should prove useful. I would suggest that these be worked up and placed in THE GUIDE. For example, plots could be made of 1000, 2000, 5000 and 10,000 fpm for each of the classifications listed below.

Perfectly smooth pipe	= 0.000005
New steel pipe	= 0.00015
Galvanized iron	= 0.0005
Average concrete	= 0.003
Average riveted steel	= 0.01

One more point requires attention and that is the correction for density. For ordinary ventilating work, we can vary friction directly as the density without serious error. For temperatures or viscosities much different from standard, it will be safest

to go back to the Moody-Rouse data. However, approximations can be made as follows:

For perfectly smooth ducts—

$$H_a = H_s \left(\frac{\rho_a}{\rho_s}\right)^{0.81} \left(\frac{\mu_a}{\mu_s}\right)^{0.19}$$

For average galvanized ducts—

$$H_a = H_s \left(\frac{\rho_a}{\rho_s}\right)^{0.90} \left(\frac{\mu_a}{\mu_s}\right)^{0.1}$$

And for very rough ducts—

$$H_a = H_s \left(\frac{\rho_a}{\rho_s}\right)$$

where

$$\left.\begin{aligned} H &= \text{friction loss} \\ \rho &= \text{density} \\ \mu &= \text{viscosity (absolute)} \end{aligned}\right\} \text{all in any consistent units}$$

and subscript a refers to actual conditions and subscript s refers to standard conditions

AUTHOR'S CLOSURE: The author would like to thank Mr. Madison for his comments and suggestions. His investigation of the round-to-rectangular conversion will permit the new chart to be used with the present equivalence factors until the Laboratory can study the problem in greater detail. His suggestion for a graph-type correction for roughness is worth further investigation and is probably the simplest answer to the difficulty mentioned in the paper. If Mr. Madison's correction equations for density are altered to permit the use of kinematic viscosity rather than absolute viscosity (since the former is given conveniently by Fig. 9 of the paper), they become simpler as follows:

For perfectly smooth ducts—

$$H_a = H_s \frac{\rho_a}{\rho_s} \left(\frac{\nu_a}{\nu_s}\right)^{0.19}$$

For average galvanized ducts—

$$H_a = H_s \frac{\rho_a}{\rho_s} \left(\frac{\nu_a}{\nu_s}\right)^{0.10}$$

For very rough ducts, the equation is not changed.

1281

No. 1281

THE TRANSMISSION OF SOLAR RADIATION THROUGH FLAT GLASS UNDER SUMMER CONDITIONS

By George V. Parmelee,* Cleveland

This paper is the result of research carried on by the American Society of Heating and Ventilating Engineers at its Research Laboratory located at Cleveland, Ohio.

INTRODUCTION

ANALYSES of available data on solar radiation, indicate that the information on the transmission of solar heat through glass and the effect of temperature difference between inside and outside is incomplete, especially with regard to the effects of shading devices, of double windows and of the use of heat absorbing glass.

Because of the great importance of securing accurate data on solar radiation transmission through glass areas in connection with calculations of cooling loads (and sometimes of heating loads), a comprehensive research program was initiated early in 1945 as recommended by the Technical Advisory Committee on Glass.† This report, setting out the methods used in the analysis and summarizing the results should be considered as a progress report on this phase of the Society's research work.

Purpose and Scope of Report

The purpose of this report is to present the results of an analysis of heat flow through glass windows exposed to solar radiation. Details of the methods by which they were obtained are given in three Appendices. As no new data are presented, confirmation by actual tests will be lacking until experimental work now in progress is completed. Most of the fundamental data required for the analysis were, however, found in the literature and only required organization in a form suitable for use by the air conditioning engineer. Although part of the data given in this paper applies only to summer conditions, it is planned to extend the analysis in a future report to cover winter conditions.

Only unshaded, unfigured, smooth, flat glass has been considered at this time. At present there is a lack of exact data on the absorption and reflection characteristics of the various types and colors of surfaces such as found in shades,

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particularly in regard to the variation of these characteristics with respect to the angle of incidence of the sun. It, therefore, did not appear practicable to attempt to evaluate shading effects mathematically.

Considerable impetus was given this preliminary analysis by the interest shown by, and the suggestions received from, the technical departments of several glass manufacturers and by the published work of Hottel and Woertz [1] on the characteristics of flat plate solar heat collectors, from which some valuable references were secured.

PHYSICAL AND CHEMICAL PROPERTIES OF GLASS

The important properties of glass to be considered are the index of refraction and the absorptivity of the glass for radiation from sources at different temperatures. Of secondary importance are specific heat, specific weight and thermal conductivity. The first two characteristics are a function of the wavelength of the incident radiation. One wavelength range to be considered lies between 0.29 μ (microns) and about 2.5 μ and includes the solar and sky spectra. In this band lie the ultraviolet, the visible and the infra-red radiations. The second range consists only of the invisible longer wavelength infrared radiation between about 4 μ and 100 μ. This range is characteristic of radiating bodies having temperatures in the neighborhood of 0 to 100 F. Ordinary glass transmits the shorter wavelengths with relative ease, whereas it is entirely opaque to the longer wavelength range. Only the effect of the former need be considered, therefore, with respect to absorption and index of refraction. It will be shown in a later section that for all practical purposes both of these properties can be considered to remain constant throughout the wavelength range of the solar spectrum.

The index of refraction is a measure of the degree to which a radiant ray is bent toward the perpendicular to the surface of a substance of high density, through which it is to pass, from a substance of lesser density such as air. A sodium light ray is often used in determining this property. Reflection of radiant waves from either surface of the substance is closely related to this property. The index of refraction μ is defined as the ratio of the sine of the angle of incidence, i, to the sine of the angle of the refracted ray, i', or

$$\text{index of refraction, } \mu = \frac{\sin i}{\sin i'} \quad \ldots\ldots\ldots \quad (1)$$

These angles are measured from the normal to the surface as shown by Fig. 8 in Appendix I. The ray after emerging from the lower surface is bent back parallel to its entering path but, of course, is displaced. An index of 1.526 was used in all calculated data given in this paper (see Appendix III).

The absorption characteristics of glass are dependent upon the chemical constituents. Ferric and ferrous oxides are two compounds used to increase the absorption of glass for solar radiation. The former is strongly absorbent in the ultra-violet portion of the spectrum, the latter in the infra-red portion and

[1] Exponent numerals refer to Bibliography.

various degrees of absorption are obtained by controlling the quantities of these two compounds. The thickness of the glass sheet is also a factor in reducing the energy of the emerging ray. Since the energy in the solar spectrum at earth level is divided among ultra-violet, visible and infra-red radiation about as 3, 44, and 53 per cent, respectively, theoretically about 56 per cent of the total energy could be excluded by a heat absorbing glass without impairing the light transmitting qualities.

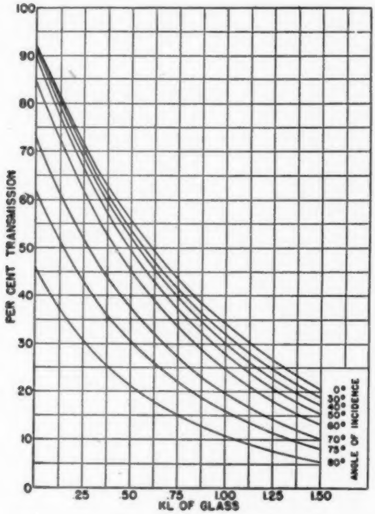

FIG. 1. TRANSMISSION OF DIRECT SOLAR RADIATION BY A SINGLE SHEET OF GLASS

Specific heat and density of the glass are related to the storage effect. In ordinary glass installations this effect is unimportant and a later section deals with calculations showing the magnitude of heat storage.

Specific thermal conductivity of glass is not of great importance in conduction heat transfer because, although the value is low compared to metals, it is high compared to the conductance of air films on either side of the glass sheet. Also, the sheets of glass in normal usage are thin compared to the thickness of other building materials, which have much lower conductivities.

To simplify the analysis of the problem, the effects of solar radiation and of temperature difference were considered separately. Direct radiation and sky radiation, the two components of solar radiation, were also treated separately. Direct radiation refers to the radiant energy in the beam or ray from the sun itself. Sky radiation refers to the short-wave radiation of the sun that has been scattered by the atmosphere and comes to the earth from all directions. It does not include the low temperature long-wave radiation from the atmospheric water vapor.

The fractions of the direct and sky radiations transmitted by glass were calculated first from its known physical properties. Then, using the same data, calculations were made of the fractions of the radiation absorbed and reflected by glass. The methods used are described in detail in Appendix I. Finally, a heat balance was set up between the absorbed solar energy and the heat transfer from the outdoor surroundings to the glass on one hand and the heat transfer from the glass to the indoor surroundings on the other hand. From this heat balance, the net flow to the interior could be computed.

Low temperature radiation and natural convection effects on the indoor side were computed separately as outlined in Appendix II. Low temperature radiation and the effect of wind on the weather side were combined in a single coefficient because of the lack of specific data on each of these points. This is discussed in greater detail in Appendix III.

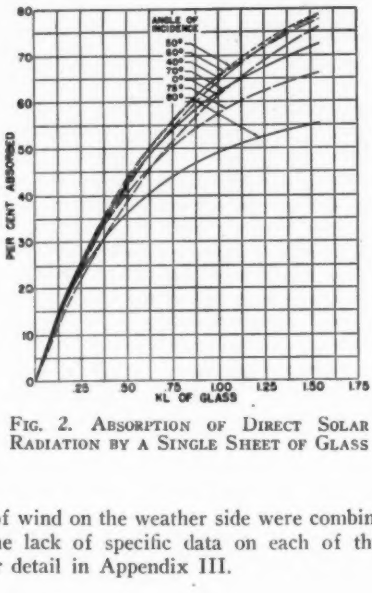

FIG. 2. ABSORPTION OF DIRECT SOLAR RADIATION BY A SINGLE SHEET OF GLASS

THE TRANSMISSION, ABSORPTION AND REFLECTION OF DIRECT SOLAR RADIATION

The following formulae, which express the transmitted, absorbed and reflected energy as fractions of the incident energy, were developed by methods outlined in Appendix I. T_1, A_1 and R_1 represent these fractions, respectively. For single sheets of glass

$$T_1 = \frac{(1 - r)^2 g}{1 - r^2 g^2} \quad \ldots \quad (2)$$

$$A_1 = 1 - r - \frac{(1 - r)^2 g}{1 - rg} \quad \ldots \quad (3)$$

$$R_1 = r + \frac{rg^2(1 - r)^2}{1 - r^2g^2} \quad \ldots \quad (4)$$

Evaluation of the right hand member of each equation required two separate computations, one for each value of r, according to Fresnel's laws of reflection. T_1, A_1 and R_1 are found by averaging these two results arithmetically. The g term expresses the depletion of the radiant energy by absorption. Both r and g are functions of the angle of incidence and are completely treated in Appendix I.

Figs. 1 and 2 show the transmission and the absorption expressed in percentages as functions of both the angle of incidence of the solar ray and the

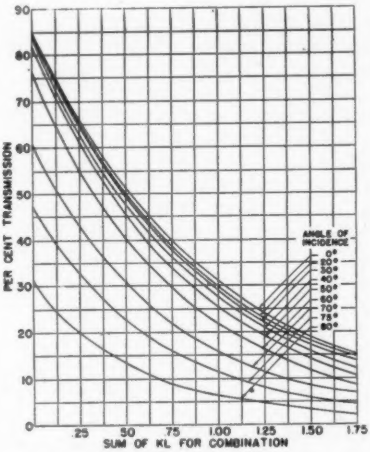

FIG. 3. TRANSMISSION OF DIRECT SOLAR RADIATION BY TWO SHEETS OF GLASS VS. THE SUM OF THE KL FOR EACH SHEET

absorption characteristic, KL, of the glass. The latter is the product of the extinction or absorption coefficient, K, and the thickness of the glass, L. The term is dimensionless, since K is expressed as absorption per inch and L is in inches [The reader should not confuse K with thermal conductivity, k, which has dimensions Btu per (hr) (sq ft) deg F per in.]. For example, the direct solar energy transmitted per unit area of the glass is the direct solar energy measured normal to the sun's rays (see Fig. 14 of Appendix III) times the cosine of the angle of incidence times the appropriate transmission percentage expressed as a fraction.

Two sheets of glass will now be considered. The subscript 1 refers to the fractions transmitted, absorbed, or reflected by the outdoor sheet as if it were a single sheet. In like manner the subscript 2 refers to the indoor sheet. The double subscript 1, 2 refers to the combination. The formulae which follow are completely developed in Appendix I.

$$T_{1,2} = \frac{T_1T_2}{1 - R_1R_2} \quad \text{(5)}$$

$$A_{1\ of\ 2} = [1 - R_2(R_1 - T_1)] \times \left[\frac{1 - (R_1 + T_1)}{1 - R_1R_2}\right] \quad \text{(6)}$$

$$A_{2\ of\ 2} = (T_1) \left[\frac{1 - (T_2 + R_2)}{1 - R_1R_2}\right] \quad \text{(7)}$$

$$R_{1,2} = R_1 + \frac{T_1^2R_2}{1 - R_1R_2} \quad \text{(8)}$$

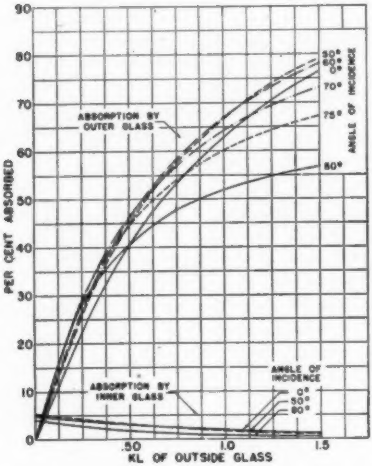

FIG. 4. ABSORPTION OF DIRECT SOLAR RADIATION BY TWO SHEETS OF GLASS WHEN THE KL OF THE INNER SHEET EQUALS 0.054

The fraction absorbed by each sheet is given, rather than the total absorption, because the computations require that each be known. The total fraction of the incident radiation that is absorbed is simply the sum of Equations (6) and (7). Fig. 3 shows the transmission of double glass expressed as a function of the angle of incidence and the sum of the KL characteristics of each of the two sheets. The order in which the sheets are placed, if they be of different characteristics, has no bearing on either the fraction transmitted or the fraction absorbed *by the combination*. However, the fraction absorbed by each sheet *and, therefore, the rate of heat transfer from the inner sheet,* depends upon the order. Fig. 4 gives these fractions for one particular case, that in which the inner glass has a KL of 0.054 (closely equal to that of grade A double strength glass). The outer glass may have any characteristic.

TRANSMISSION, ABSORPTION AND REFLECTION OF SKY RADIATION

Although the intensity of the short wave radiation from the sky is relatively low as compared to the direct intensity of the sun, its effect on heat gains in total may be considerable. For example, buildings are often arranged with large window areas exposed to the northern sky. The heat flow in such cases may amount to a sizable fraction of the total cooling load.

Because the sky radiation comes from all angles, it was necessary to compute the fractions transmitted, absorbed and reflected by methods of graphical integration outlined in Appendix I. In effect the radiation can be considered

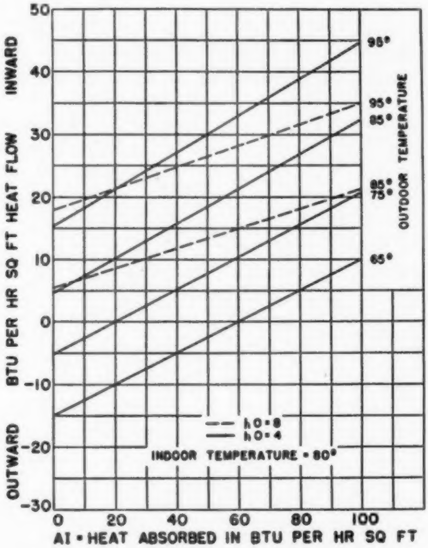

FIG. 5. HEAT FLOW AT THE INDOOR SURFACE OF A SINGLE SHEET OF GLASS

TABLE 1—FRACTIONS OF SKY RADIATION TRANSMITTED (T), ABSORBED (A_1 AND A_2), AND REFLECTED (R) BY SINGLE AND DOUBLE GLASS (INDEX OF REFRACTION = 1.526)

	T	A_1	A_2	R
Single glass (KL = 0.054)	0.778	0.060	...	0.162
Single glass (KL = 0.419)	0.510	0.365	...	0.125
Double glass (both sheets KL = 0.054)	0.631	0.067	0.048	0.254
Double glass (inside KL = 0.054) (outside KL = 0.419)	0.411	0.393	0.032	0.164
Double glass (both sheets KL = 0.419)	0.269	0.387	0.190	0.154

as coming from an angle of about 60 deg measured from the perpendicular to the glass. The results for certain types and combinations of glass are given in Table 1.

HEAT FLOW DUE TO ABSORBED HEAT AND TEMPERATURE DIFFERENCE

Fig. 5, the calculations for which are explained in detail in Appendices II and III, shows the rate of heat flow from a single sheet of glass to the inte-

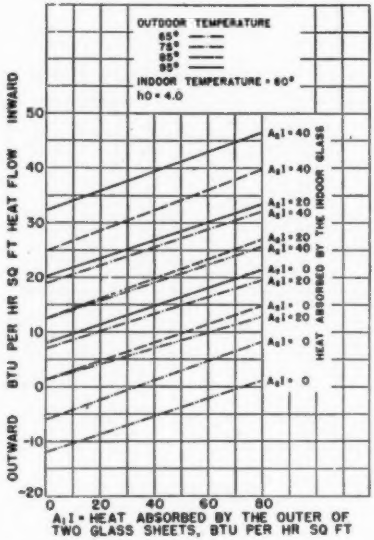

FIG. 6. HEAT FLOW AT THE INDOOR SURFACE OF THE INNER OF TWO SHEETS OF GLASS

rior (or the outside, depending upon the temperature). This heat flow depends upon three factors—the difference between inside and outside air temperatures, the fractions of the direct and sky radiations absorbed and the outside film coefficient. Fig. 3 and Table 1 are used to estimate the solar heat absorbed. It also depends to some extent upon the mean temperature, since this affects the unit rate of low temperature radiation from the glass to the indoor surroundings. Four curves are drawn for several outdoor temperatures and an outdoor film coefficient, h_o, of 4.0. Both convection and low temperature radiation are combined in the coefficient because there is at present insufficient experimental data to consider each separately. Further, it is assumed that the indoor surroundings seen by the glass are at the same temperature as the indoor air. The data given in Fig. 5 would not apply to a structure,

for example, a greenhouse, in which most of the surface is glass and would therefore be at the same temperature. Heat gains or losses by radiation obviously cannot take place between surfaces at the same temperature.

Although the curves are not exactly straight lines, they were so drawn. They are seen to diverge to some extent, indicating that the fraction of the absorbed solar heat transmitted to the interior is not constant. The divergence is also due to changes in heat transfer coefficients because of temperature changes. For zero heat absorption, the overall transmission coefficient can be estimated and, in spite of some variations, is substantially equal to 1.0 Btu per (hr) (sq ft) (deg F) for the conditions assumed.

The curves, although drawn for an 80 F indoor temperature, are applicable to any indoor temperature within 5 to 10 deg of 80 F. For example, if the indoor temperature were 70 F and the outdoor 85 F, the temperature difference would be 15 deg and the curve for 95 F outside, 80 F inside, could be used. Two curves for $h_o = 8.0$ are shown to illustrate the effect of increasing the convection on the outer surface of the glass.

Fig. 6 gives data applicable to double glass. Here it is necessary to know the amount of heat absorbed by each sheet of glass. Fig. 4 and Table 1 have been produced for that purpose.

METHOD OF CALCULATING HEAT GAINS DUE TO SOLAR RADIATION ON GLASS

The curves presented in this report together with other data given in the Appendices permit the calculation of heat gains due to solar radiation on unshaded windows. Their use is best illustrated by examples. Let it be assumed that the indoor temperature to be maintained is 80 F, that the outdoor temperature is 95 F, that the sun time is 2 p.m. and that the date is July 1. The window, which lies flush with the surface of a western wall, is single and has a *KL* of 0.054. The latitude is 40 deg north. It will be assumed that curve A of Fig. 14 (Appendix III) is applicable.

Solution:

From Fig. 16 (Appendix III) the altitude is 62 deg; the angle between the sun and the surface is 26 deg. Hence the angle of incidence is 90—26 or 64 deg.

From Fig. 14 (Appendix III) the direct normal solar radiation is 283 Btu per (hr) (sq ft). The radiation on the window is 283 x cos 64 deg or 124 Btu per (hr) (sq ft).

Fig. 15 (Appendix III) Curve A (recommended in the absence of more complete data) shows that the direct solar radiation at 62 deg altitude is 5.3 times the sky radiation. Therefore, the sky radiation on a horizontal plane is (283 ÷ 5.3) (sin 62 deg). Half of this or 23.5 falls on a vertical plane.

From Fig. 1, 73 per cent of the direct energy, or 90.5 Btu per (hr) (sq ft) is transmitted. From Table 1, 77.8 per cent of the sky radiation, or 18.5 Btu, is transmitted; 6 per cent, or about 1.5 Btu, is absorbed.

From Fig. 2, 6 per cent of the direct radiation, or about 7.5 Btu, is absorbed, making a total of 9 Btu per (hr) (sq ft) absorbed.

In Fig. 5 the curves for $h_o = 4.0$ are recommended for summer use. For a 95 F outdoor temperature, an 80 F indoor temperature and 9 Btu solar heat absorbed, the heat flow inward is 18 Btu. The total heat flow is summarized as follows:

Direct solar radiation transmitted	90.5
Sky radiation transmitted	18.5
Heat flow due to absorption and temperature difference	18.0
Total, Btu per (hr) (sq ft)	127.0

If a heat absorbing glass were added on the weather side of the plain glass, the heat transmission would be revised as follows:

From Fig. 3 for a *KL* sum of 0.473, the percentage transmitted is 37 per cent, or 46 Btu. Fig. 4 shows that the outer glass absorbs 41 per cent or 51 Btu; the inner glass absorbs 2.5 per cent or about 3 Btu. Table 1 shows that 41.1 per cent of the sky radiation is transmitted, or about 9.5 Btu; 39.3 per cent absorbed by the outer glass, or 9.5 Btu; and 3.2 per cent absorbed in the inner, or about 1.0 Btu. The total absorption is 60.5 Btu by the outer sheet and 4 Btu by the inner. The heat flow from Fig. 6 for this combination is thus 20.5 Btu per (hr) (sq ft).

Summarizing:

Direct solar radiation transmitted	46.0
Sky radiation transmitted	9.5
Heat flow due to absorption and temperature difference	20.5
Total, Btu per (hr) (sq ft)	76.0

The calculations shown are somewhat tedious. However, when a systematic study has been made of the characteristics of typical glasses, it will be possible to express the total heat flow for any particular glass as a function of the radiation intensity, the angle of incidence and the temperature difference.

It must be clearly understood by the reader that experimental data which wholly substantiate these calculations have not yet been obtained. Further, because of the storage capacity of interior walls and floors on which will fall the sunlight which passes through a window, some lag must be expected between the time at which the solar heat passes through the glass and the time at which the heat load influences the cooling system. Heat storage characteristics of the glass itself are discussed in Appendix III.

ACKNOWLEDGMENTS

The data in this paper are the result of many hours of painstaking work. The writer is grateful for the assistance of Doris M. Dietz and George Springer, research assistants, who made the computations. Considerable labor was saved by Mr. Springer's mathematical treatment of the various problems. Thanks are due the technical departments of the Libbey-Owens-Ford Co., Mississippi Glass Co. and Pittsburgh Plate Glass Co. for data which appear in this paper and the chairman and members of the Technical Advisory Committee on Glass for helpful criticisms and suggestions. The writer also wishes to thank Cyril Tasker, director of research, for his encouragement and criticisms and for his suggestions in regard to the arrangement of the paper which, it is hoped, will stimulate reader interest in research papers.

APPENDIX I

THE CALCULATION OF TRANSMISSION, ABSORPTION AND REFLECTION OF SOLAR ENERGY BY GLASS

Single Glass

If one looks out through a window at night from a lighted room one sees two reflected images of objects in the room. Closer examination would reveal others of much diminished intensity. The two images are reflections from the first and second

surfaces of the glass; the others are due to successive internal reflections. Stokes [2] in 1862 analyzed mathematically this effect of multiple reflections of light incident upon one or more sheets of a transparent substance. The general method used by Stokes has been followed in this paper but with considerable simplification in the treatment of several sheets of glass separated by air spaces.

Light and heat radiation are considered to be electromagnetic transversely vibrating waves, components of which vibrate in different planes. The reflection of any single component will depend upon the plane in which it is vibrating. This radiant energy

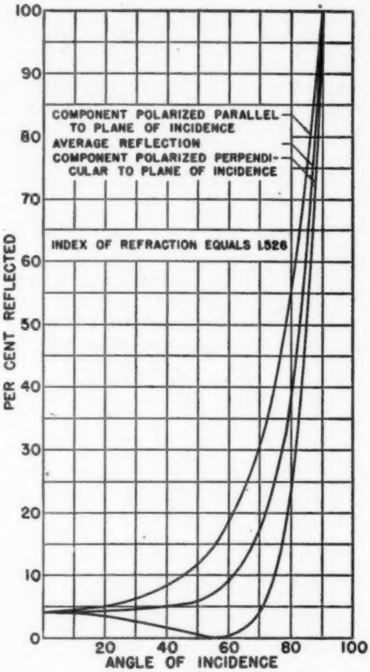

FIG. 7. REFLECTION OF DIRECT SOLAR RADIATION FROM A SINGLE SURFACE AS A FUNCTION OF ANGLE OF INCIDENCE

will be considered as having components of equal intensity in all planes. Each such component may be broken down into two, one vibrating in a plane perpendicular to the sheet of glass, the other in a plane parallel to the glass. If these two sub-components are summed, the total radiation can then be considered as consisting of two components, one in each plane. Fresnel has expressed the specular reflection of each of these components from a single surface by the following equations:

$$r_{pl} = \frac{\sin^2 (i - i')}{\sin^2 (i + i')} \quad (9)$$

$$r_{pd} = \frac{\tan^2 (i - i')}{\tan^2 (i + i')} \quad (10)$$

where

i = angle of incidence.
i' = angle of refraction.
r_{pl} = fraction reflected of the component parallel to the plane of incidence.
r_{pd} = fraction reflected of the component perpendicular to the plane of incidence.

At normal incidence, the amount of each component reflected is the same, and if air and glass are the mediums involved, the fraction reflected is expressed by the formula

$$r = \frac{(\mu - 1)^2}{(\mu + 1)^2} \qquad (11)$$

where μ is the index of refraction of the glass. Fig. 7 shows the percentage of each component reflected from a single surface, as well as the average.

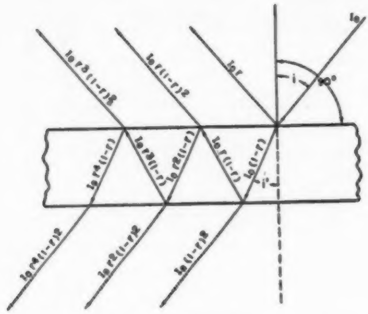

Fig. 8. Multiple Reflections of Direct Solar Radiation from a Single Sheet of Glass

Fig. 8 represents the first, second and several successive internal reflections of a ray striking the surface of a single sheet of glass at the angle of incidence i. The intensity of radiation is designated by I_o. First, let it be assumed that there is no absorption by the glass. The total energy transmitted, I, is given by the sum of the infinite series:

$$I = I_o\,[(1-r)^2 + r^2\,(1-r)^2 + r^4\,(1-r)^2 + \ldots\ldots] \qquad (12)$$

where r is the fraction of each component reflected calculated by Equations (9), (10), or (11). Since the incident energy has been assumed to be equally divided between the two components, the total fraction transmitted, T', neglecting absorption, is, by summation of the series

$$T' = \frac{I}{I_o} = \frac{T'_{pl} + T'_{pd}}{2} \qquad (13)$$

where

$$T'_{pl\ (or\ pd)} = \Sigma_{pl\ (or\ pd)}\ (1-r)^2 + r^2\,(1-r)^2 + r^4\,(1-r)^2 + \ldots\ldots$$

$$T'_{pl\ (or\ pd)} = \frac{1 - r_{pl}}{1 + r_{pl}} \text{ or, } \left[\frac{1 - r_{pd}}{1 + r_{pd}}\right] \qquad (14)$$

However, as each ray passes through the glass sheet part of the energy is absorbed. If the sheet of glass be divided into a number of layers of equal thickness, dL, each

of which reduces the intensity by dI in proportion to its thickness, dL, and its absorption or extinction coefficient, K, then

$$-dI = IKdL \qquad (15)$$

Integrating between limits of I_o and I and zero and L:

$$\frac{I}{I_o} = e^{-KL} = g' \qquad (16a)$$

where g' equals the fraction of the energy available after passage through the glass. The thickness of the glass, L, must, of course, be modified to account for the refraction or bending of the ray toward the perpendicular. The KL term is modified, then, by dividing by

$$\sqrt{1 - \frac{(\sin i)^2}{(\mu)^2}}$$

so that the modified fraction available, g, becomes

$$\log_e g = \frac{-KL}{\sqrt{1 - \frac{(\sin i)^2}{(\mu)^2}}} \qquad (16b)$$

The term K, which has units per inch, should not be confused with specific thermal conductivity, k, which has units Btu per (hr) (sq ft) (deg F per inch). The infinite series given by (12) when modified by g becomes

$$I = I_o\,[g\,(1-r)^2 + r^2g^3\,(1-r)^2 + r^4g^5\,(1-r)^2 + \ldots\ldots] \qquad (17)$$

The general expression for the transmission of any component, then, becomes, for single glass sheets:

$$T_1 = \frac{(1-r)^2 g}{1 - r^2g^2} \qquad (18)$$

Similarly the expression for reflection is found to be:

$$R_1 = r + \frac{rg^2\,(1-r)^2}{1 - r^2g^2} \qquad (19)$$

And since the fraction absorbed by the glass, A_1, is found thus:

$$A_1 = 1 - R_1 - T_1 \qquad (20)$$

the expression for absorption is:

$$A_1 = 1 - r - \frac{(1-r)^2 g}{1 - rg} \qquad (21)$$

The total transmission, reflection and absorption are found by evaluating Equations (18), (19) and (21) respectively, for each component and averaging. It can be seen from Fig. 8 that if the reflections beyond the second order are neglected, the transmission can also be expressed as follows:

$$T_1 = g\,(1-r)^2 \qquad (22a)$$

This amounts to neglecting the value of r^2g^2 in the denominator of Equation (18) and, as long as r is small, the error involved is negligible except at high angles of incidence as shown in Table 2. Inspection of Equation (12) shows that another approximation is possible if the g term is omitted from the series given by Equation (17) and then introduced as follows:

$$T_1 = \frac{g(1-r)}{(1+r)} \qquad (22b)$$

TABLE 2—COMPARISON OF THE TRANSMISSION OF SINGLE GLASS CALCULATED BY THREE METHODS ($\mu = 1.526$)

Angle of Incidence	$KL = 0.054$			$KL = 0.837$		
	Eq (18)	Eq (22a)	Eq (22b)	Eq (18)	Eq (22a)	Eq (22b)
0 deg	0.868	0.883	0.877	0.396	0.404	0.401
40 deg	0.854	0.851	0.854	0.359	0.359	0.360
80 deg	0.415	0.362	0.426	0.134	0.130	0.153

Again only a small error is involved, as Table 2 shows. The error is greatest for glasses of high transmission value and at high angles of incidence.

Similar approximations can be made for the fraction of the incident solar radiation absorbed by the single glass. From Equation (20) it follows that absorption may be expressed approximately as:

$$A_1 = (1 - r)^2 - g\,(1 - r)^2 \qquad (23a)$$

or

$$A_1 = \frac{(1 - r)}{(1 + r)} - g\,\frac{(1 - r)}{(1 + r)} \qquad (23b)$$

or

$$A_1 = (1 - g) \qquad (23c)$$

The degree of approximation of these three expressions is shown by Table 3, where the error is seen to be considerable. However, in the high transmission glasses there is so little absorption that the error is negligible except at high angles of incidence.

Double Glass

The method used in the preceding section can be used to calculate the transmission and absorption of two air spaced sheets of glass. The subscript 1 designates the outer glass, the subscript 2 designates the inner glass; the double subscript represents the combination. The series is set up in the following manner. The second sheet transmits T_2 times that transmitted by the first sheet or T_1T_2. The energy reflected by the second sheet followed by a re-reflection from the first sheet and transmission by the second sheet is $T_1R_2R_1T_2$. The next term is $T_2R^2{}_2T_1R^2{}_1$ and so on, so that

$$T_{1,2} = T_2T_1 + T_2T_1R_1R_2 + T_2T_1R^2{}_1R^2{}_2 + \ldots \qquad (24)$$

It follows then that

$$T_{1,2} = \frac{T_1T_2}{1 - R_1R_2} \qquad (25)$$

TABLE 3—COMPARISON OF THE ABSORPTION OF SINGLE GLASS CALCULATED BY FOUR METHODS ($\mu = 1.526$; $KL = 0.837$)

Angle of Incidence	Eq (21)	Eq (23a)	Eq (23b)	Eq (23c)
0 deg	0.553	0.529	0.515	0.567
40 deg	0.584	0.544	0.547	0.603
80 deg	0.461	0.258	0.304	0.666

$$R_{1,2} = R_1 + \frac{T^2_1 R_2}{1 - R_1 R_2} \qquad (26)$$

$$A_{1,2} = \frac{[1 - (R_1 + T_1)][1 - R_2(R_1 - T_1)] + T_1[1 - (T_2 + R_2)]}{1 - R_1 R_2} \qquad (27)$$

However, in working out heat balances it is necessary to find the absorption of each glass separately. Hence,

$$A_{1\ \mathrm{of}\ 2} = \frac{[1 - (R_1 + T_1)][1 - R_2(R_1 - T_1)]}{1 - R_1 R_2} \qquad (28)$$

$$A_{2\ \mathrm{of}\ 2} = \frac{[1 - (T_2 + R_2)]\,T_1}{1 - R_1 R_2} \qquad (29)$$

R, *T* and *A* are calculated for single sheets as before and substituted in these equations. Inspection of Equation (25) shows that $T_{1,2}$ is expressed approximately as

$$T_{1,2} = T_1 T_2 \qquad (30)$$

This simplification breaks down when R_1 and R_2 become reasonably large, as they do at high angles, as shown by Table 4.

TABLE 4—COMPARISON OF THE TRANSMISSION OF DOUBLE GLASS CALCULATED BY TWO METHODS (μ = 1.526; BOTH SHEETS THE SAME)

ANGLE OF INCIDENCE	KL = 0.054 (ONE SHEET)		KL = 0.419 (ONE SHEET)	
	Eq (25)	Eq (30)	Eq (25)	Eq (30)
0 deg	0.756	0.754	0.365	0.362
40 deg	0.740	0.729	0.305	0.297
80 deg	0.250	0.172	0.141	0.117

Again the error is greatest for glasses of high transparency and for incident radiation at the higher angles, where the R_1R_2 term in Equation (25) becomes appreciable. Similar approximations can be made for the absorption of energy by either the inner or outer sheet. Throughout this paper the exact equations, which require only one additional step, have been used.

Transmission and Absorption of Typical Glass

Since the primary purpose of this paper was to obtain data that could be useful in correlating test work with theory, Figs. 1 through 6 were prepared and have been presented in the main part of this report. It was necessary to examine typical glasses in order to determine both the index of refraction of window glass and the range of *KL* values that might obtain in practice.

Morey [16] gives the composition of a great variety of glasses. For window glass made by present day processes, the constituents are found in approximately the following percentages:

SiO_2	Na_2O	CaO	MgO	Al_2O_3	Fe_2O_3
70 to 73	12 to 15	9 to 14	0 to 3	0 to 1.5	0 to 0.15

For glasses having percentage compositions in the ranges given, Morey gives values of the refractive index at the *D* line (sodium light) of between 1.502 and 1.549. Since the technical departments of the glass companies usually take the index

as being 1.526, this value has been used in all calculations. However, the possible effect of variations has been studied and the results given in Appendix III.

The KL values were obtained by computation from transmission data for both plain and heat absorbing glasses furnished by a number of manufacturers. These data gave either the spectral transmission curves, or the transmission computed from this curve, or both, for normally incident radiation. Other data were obtained from the recent paper of Hottel and Woertz [1] and the earlier paper of Houghten, Blackshaw and Gutberlet [12]. The transmission data given in reference 12 was computed by subtracting tabular values of percentages intercepted from 100. In this paper [12] the interception was attributed to absorption, but later studies at the A.S.H.V.E. Laboratory [7] showed that reflection was chiefly responsible. The transmission data, however, are undoubtedly representative of the samples tested.

TABLE 5—CHARACTERISTICS OF SOME GLASS SAMPLES FOR NORMALLY INCIDENT SOLAR RADIATION

GLASS TYPE	THICKNESS L, INCHES	PER CENT TRANSMISSION	g	KL	ABSORPTION COEF K	REF.
Single strength A quality	0.071	88.8	0.971	0.030	0.411	12
Double strength A quality	0.127	89.2	0.976	0.025	0.194	12
Clear plate	0.270	87.2	0.954	0.047	0.174	12
Grade A	...	...	...	...	0.432	1
Water white	...	...	...	...	0.107	1
Tank plate	0.243	77.5	0.848	0.166	0.683	13
Water white	0.259	88.4	0.967	0.033	0.129	13
Heat absorbing, H_1	0.254	39.5	0.432	0.840	3.300	13
Heat absorbing, H_2	0.129	43.5	0.476	0.742	5.750	13
Heat absorbing, H_3	0.111	51.0	0.558	0.584	5.260	14
Heat absorbing, H_4	0.247	34.0	0.373	0.986	3.990	14
Heat absorbing, H_6	0.232	18.5	0.202	1.598	6.890	18

To date no systematic study has been made of standard grades and thicknesses of commercial window glass to determine the variability of their transmission characteristics. This is a part of the research project now in progress.

Table 5 gives transmission and absorption data from the sources just mentioned and listed in the last column. The numbers apply to bibliographic reference.

Except for the data ascribed to Hottel and Woertz,[1] the g values were computed from data giving the transmission at normal incidence. Hottel and Woertz determined K by measuring the normal transmission of one or more plates of the same glass, cemented together to avoid reflection at the interfaces. Their paper refers to the variability of characteristics of pieces of glass of presumably the same grade and batch. Table 5 gives further evidence of that fact.

The g value of glass can be obtained by rearranging Equation (18) and solving for g as follows:

$$g = \frac{1}{2T}\left[- \left(\frac{1-r}{r}\right)^2 + \sqrt{\left[\frac{1-r}{r}\right]^4 + \frac{4T^2}{r^2}} \right] \quad \text{(31)}$$

This equation is awkward to use but if the second term inside the brackets be expanded by the binomial theorem, the first order approximation is accurate within about 0.1 per cent for normal incidence. Therefore,

$$g = T\left(\frac{1}{1-r}\right)^2 \quad \text{(32)}$$

This, of course, is simply Equation (22*a*) rearranged. For angles other than normal incidence $\left(\frac{1}{1-r}\right)^2$ must be evaluated for each component of polarization and the arithmetical average determined. For normal incidence and an index of refraction of 1.526

$$g = 1.094T \quad \ldots \ldots \quad (33)$$

The results of this part of the analysis have been given in the form of curves in the main portion of the report. However, some discussion of them is appropriate here. Fig. 1 gives transmission values for single glass having an average index of

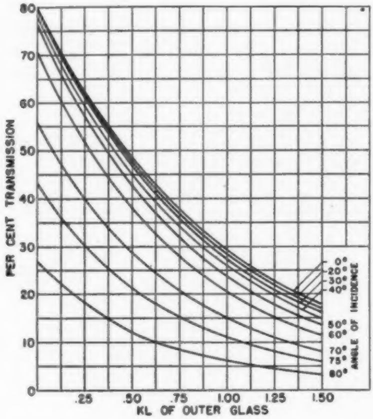

FIG. 9. TRANSMISSION OF DIRECT SOLAR RADIATION BY TWO SHEETS OF GLASS WHEN THE KL OF THE INNER SHEET EQUALS 0.054

refraction of 1.526 for KL values ranging from zero to 1.55. This adequately covers the practical range. Fig. 9 gives transmission values for a double glass combination in which the inner glass is double strength plain window glass of KL equal to 0.054. The outer glass may be any type ranging from $KL = 0.0$ to $KL = 1.55$. Fig. 3 gives similar values for double windows but with the abscissa as the sum of the individual KL values of each sheet. Actually the transmission of two windows requires separate calculations for each combination although the transmission is independent of the order in which they come. However, comparison of Figs. 3 and 9 shows that the error is negligible if the KL of each are summed and Fig. 3 is used.

In order to calculate the heat flow from the glass to the interior, it is necessary to know the glass temperature. This is a function of the indoor and outdoor temperatures and the absorption characteristics of the glass. Fig. 2 shows absorption plotted as a function of KL with angle of incidence as the parameter. It will be seen that absorption is substantially constant with incident angle for small KL values but that there is considerable change as KL increases. The percentage absorption for any type of glass is seen to be practically constant between 40 and 60 deg incident angle.

As stated before, the total absorption of a double glass combination is independent of the order in which the sun strikes the sheets. This is not true of the percentage absorbed by each. Presentation of the data becomes voluminous unless limitations

are placed on the combinations. Fig. 4 shows solar heat absorption for various types of glass outdoors with one-eighth inch double strength plain glass (KL value of 0.054) indoors.

Sky Radiation, Transmission, Absorption and Reflection

In order to facilitate the analysis, the following assumptions were made: (1) that the radiation is uniform from all quarters of the sky and is independent of the position of the sun and (2) that the polarization of the sky radiation is negligibly small.

Fig. 10 represents a window viewing one-quarter of the sky. Because the area of the window is small compared to that of the sky, i may represent the angle of incidence between the radiant ray from the unit sky area, dA_2, to any part of the window.

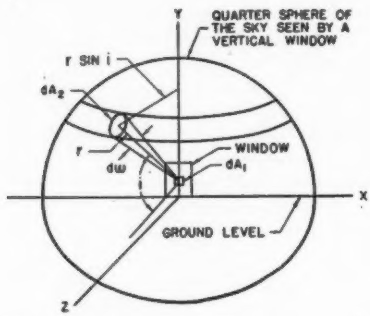

FIG. 10. GEOMETRY OF SKY RADIATION ON A VERTICAL WINDOW

If I_{os} represents the unit intensity of the sky radiation, the energy, dq, received by the window is proportional to the solid angle, dw, and the cosine of the angle of incidence, hence

$$dq = I_{os}\, dA_1 \cos i\, dw \text{ Btu per hour} \quad (34)$$

But $\frac{dq}{dA_1}$ equals the unit energy, dE_s, received by the window from the sky and the solid angle, dw, equals $\frac{dA_2}{r^2}$. Over the whole quarter sphere at a fixed angle, i, dA_2 equals π $(r \sin i)$ (rdi). The total energy, E_s, received per unit area of window from a sky of uniform unit intensity, I_{os}, is found by integration over the quarter sphere.

The total fraction of sky radiation reflected, r_s, is the summation of the energy reflected at each angle divided by the total incident radiation. Since the fraction reflected at any one angle is a function of that angle, r (i) and the plane of polarization, the fraction of the total sky radiation reflected is

$$r_s = \int_0^{\frac{\pi}{2}} \frac{r\,(i)\,\frac{\pi I_{os}}{2} \sin\,(2\,i)\,di}{\frac{I_{os}}{2}} = \int_0^{\frac{\pi}{2}} \pi r\,(i) \sin\,(2\,i)\,di \quad (35)$$

where r (i) is expressed by Equations (9) and (10). Each expression was evaluated by plotting the integrand as a function of i and then graphically integrating

under the curve. The fraction reflected of the component polarized parallel to the plane of incidence was 0.154, which is equivalent to the reflection of a ray incident at 56 deg. The fraction of the other component reflected was 0.039, corresponding to the reflection of a ray incident at 69 deg.

The g term, found in similar fashion for two KL values, was simpler to integrate because the plot of the function was substantially a straight line. Table 1, given in the main part of this paper, lists the transmission, absorption and reflection for several glass combinations calculated by means of the r and g values found in the manner described previously, and the formulae given in Appendix I.

APPENDIX II

HEAT TRANSMISSION DUE TO ABSORPTION AND TEMPERATURE DIFFERENCE

Heat Flow at the Indoor Surface of a Single Glass Sheet

The total heat flow through a glass window is the sum of the directly transmitted solar energy and the heat transferred from the inner surface to the indoor surroundings by convection and radiation. The latter is affected by the amount of solar and sky heat absorbed by the glass. In the report, Heat Transmission Through Windows and Glass Panels,[7] the following heat balance is given for the steady state:

$$AI = h_{ro}\,(t_{so}-t_o) + h_{co}\,(t_{so}-t_{oa}) + h_{ri}\,(t_{is}-t_i) + h_{ci}\,(t_{is}-t_{ia}) \quad \ldots\ldots \quad (36)$$

where

A = the fraction of the incident radiation absorbed.
I = the incident radiation, Btu per (hr) (sq ft).
h_{ro} = outdoor radiation coefficient, Btu per (hr) (sq ft) (deg F).
h_{ri} = indoor radiation coefficient, Btu per (hr) (sq ft) (deg F).
h_{co} = outdoor convection coefficient, Btu per (hr) (sq ft) (deg F).
h_{ci} = indoor convection coefficient, Btu per (hr) (sq ft) (deg F)
t_{so} = temperature of outer surface of the glass, deg F.
t_{si} = temperature of inner surface of the glass, deg F.
t_o = temperature of the outdoor objects seen by the glass, deg F.
t_i = temperature of indoor objects seen by the glass, deg F.
t_{oa} = temperature of outdoor air, deg F.
t_{ia} = temperature of indoor air, deg F.

At this writing definite information is lacking regarding the exchange of heat by low temperature radiation between the glass and outdoor objects including the sky. Hence, h_{ro} and h_{co} will be lumped in a single coefficient, h_o and t_o will be assumed equal to t_{oa}. Further, it will be assumed that the glass temperature is uniform; thus $t_{so} = t_{si} = t_s$. It will also be assumed that $t_i = t_{ia}$. In some instances it will be convenient to combine h_{ci} and h_{ri} in a single coefficient, h_{cri}, a function both of mean temperature and temperature difference.

In arriving at the heat flow through single and double glass, an indoor temperature, t_i, of 80 F and an outdoor film coefficient, h_o, of 4.0 have been selected as being representative of average summer conditions. In the report cited,[7] it was shown that the indoor film coefficient was the controlling factor in the normal flow of heat through glass and that considerable, say 20 per cent, variation could in some instances be made in the outdoor coefficient without affecting the overall coefficient more than 3 or 4 per cent. This would not be true in the case of heat absorbing glass where the outdoor coefficient assumes more importance.

For the purposes of this paper, then, Equation (36) is written for single glass as follows:

$$AI = h_o\,(t_s-t_o) + h_{ri}\,(t_s-t_i) + h_{ci}\,(t_s-t_i) \quad \ldots\ldots\ldots\ldots \quad (37)$$

The radiation heat transfer, Q_r, between the inner surface of the glass and its surroundings is given by the following equation:

$$Q_r = 0.173\; F_e F_a \left[\frac{(T_s)^4}{(100)^4} - \frac{(T_i)^4}{(100)^4}\right] \quad \ldots\ldots\ldots\ldots \quad (38)$$

The term $0.173 \left[\frac{(T_s)^4}{(100)^4} - \frac{(T_i)^4}{(100)^4}\right]$ can be divided by $T_s - T_i$ to obtain what can be called a radiation coefficient. As shown in the report[7] referred to, this plots as a straight line function of the mean temperature, $t_m = \frac{t_s + t_i}{2}$, on semi-log paper. This is reproduced here as Fig. 11. T_s and T_i are degrees F + 460.

The angle factor, F_a, for a window in one side of a wall which sees three other walls, the floor and the ceiling, all at substantially ambient air temperature, is unity. The emissivity factor of glass has been taken at 0.94 (see McAdams[6]) and, if that of the indoor surfaces is taken as 0.85, the emissivity factor, F_e, becomes 0.80. The product of F_e times F_a times the appropriate coefficient taken from Fig. 11 gives the radiation coefficient, h_r.

It is assumed here that air circulation near the window on the indoor side is due entirely to natural means. Various authorities have correlated the heat transfer

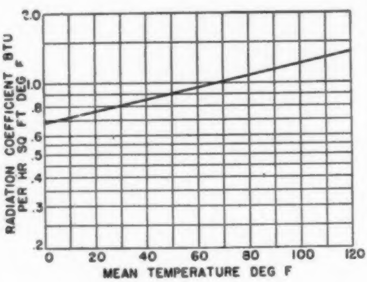

FIG. 11. RADIATION COEFFICIENT AS A FUNCTION OF MEAN TEMPERATURE FOR $F_e \times F_a = 1.0$

coefficient for natural convection by use of the difference between the surface and air temperatures raised to the 0.25 power. Hence, $h_c = C\ (t_s - t_i)^{0.25}$ where C is a constant depending upon the orientation of the surface. McAdams'[6] evaluation of C for vertical surfaces over 24 in. in height is 0.27. Colburn[3] as a result of his studies gives C a value of 0.30. In this paper McAdams' evaluation of 0.27 for convection heat transfer from a single vertical plate has been used.

Inspection of Equation (37) shows that the heat flow into the room, Q_{in}, for single glazed windows may be expressed as follows:

$$Q_{in} = AI - h_o\,(t_s - t_o) = h_{ri}\,(t_s - t_i) + C\,(t_s - t_i)^{1.25} \quad \text{. . . . . (39)}$$

This equation shows that the heat flow is a function of the mean of t_s and t_i, since the value of h_{ri} depends upon this and also the difference between t_s and t_i. Calculation of Q_{in} (or Q_{out}, depending upon the temperatures) requires a trial and error solution of Equation (39) to determine t_s. This was carried out graphically by plotting Q against t_s for different combinations of AI, h_o and t_o. The plot consisted of a series of straight lines, representing the left hand member of Equation (39) and a single curve, concave downward, representing the right hand member.

Fig. 5, given in the main portion of this paper, is a cross plot of the graphical solution of Equation (39) for an indoor temperature of 80 F and two different outdoor film coefficients, as indicated. For all practical purposes the lines are straight

and have been so drawn. From these curves the U value, when no solar heat was absorbed, ranged between 0.98 and 1.09 for indoor temperatures of 70 or 80 F, an outdoor film coefficient of 4.0 and an outdoor temperature anywhere between zero and 100 F.

It is, therefore, sufficiently accurate to express the heat flow due to temperature difference and solar heat absorption combined as a fraction of the absorbed radiation plus the heat flow at zero heat absorption.

For $h_o = 4.0$:

$$Q = 1.04\,(t_o - t_i) + 0.275\,AI \qquad (40)$$

For $h_o = 8.0$:

$$Q = 1.18\,(t_o - t_i) + 0.15\,AI \qquad (41)$$

The fraction of the absorbed heat transmitted as given by each of these equations is an average for the conditions being considered. For example, it varied between 25 and 30 per cent for $h_o = 4.0$. It is substantially equal to the average U value divided by h_o.

Heat Flow from the Indoor Surface of Two Glass Sheets

This problem is complicated by the fact that each plate absorbs a different percentage of the incident radiation. This depends upon the characteristics and the order of arrangement of the two plates, although the total absorption is independent of the order. In the following equations each sheet of glass is assumed to be at uniform temperature. The subscript 1 refers to the outer glass and the subscript 2 refers to the inner glass. Prime marks are used to designate the convection and radiation coefficients applicable to the heat exchange between the two glass sheets. Heat balances are written as follows:

$$A_1 I = h_o\,(t_{s1} - t_o) + C'\,(t_{s1} - t_{s2})^{1.25} + h'r\,(t_{s1} - t_{s2}) \qquad (42a)$$

and

$$A_2 I + C'\,(t_{s1} - t_{s2})^{1.25} + h'r\,(t_{s1} - t_{s2}) = C\,(t_{s2} - t_i)^{1.25} + h_{ri}\,(t_{s2} - t_i) \qquad (42b)$$

The heat flow into the room, disregarding the directly transmitted energy, can therefore be expressed as follows:

$$Q_{\text{in}} = C\,(t_{s2} - t_i)^{1.25} + h_{ri}\,(t_{s2} - t_i) \qquad (43)$$

$$Q_{\text{in}} = (A_1 + A_2)\,I - h_o\,(t_{s1} - t_o) \qquad (44)$$

In order to determine Q by either Equation (43) or (44), it was first necessary to determine t_{s1} and t_{s2} by methods like those used in the preceding section. It was also necessary to evaluate the constant C' for vertical air spaced plates. Present data in the HEATING, VENTILATING AIR CONDITIONING GUIDE [8] expresses the heat transfer across air spaces as a function of the mean temperature. This, however, does not take into consideration the fact that the temperature difference between the two surfaces forming the air space is an equally important factor. Rowley [4] gives results of guarded hot box tests of sheets of building board separated by air spaces of ½, 1 and 1½ in. Sufficient data were given so that the radiation contribution to the air space coefficient could be calculated and deducted. Average values of C equal to 0.211, 0.169 and 0.157 respectively were obtained. Griffith and Davis [17] found that the constant C averaged 0.19 for vertical plates spaced as indicated and decreased slightly as the space increased. Wilkes [9] gives a value of 0.18 for a 4 in. vertical space. On the basis of these data a value of C equals 0.18 was selected as being representative of air spaces likely to be encountered in practice. Fig. 6 in the main part of this report is a plot of the heat flow through double glass as a function of the amount absorbed by each sheet and the outdoor temperature. An indoor temperature of 80 F was used in these calculations.

APPENDIX III

DISCUSSION OF DATA USED IN CALCULATIONS

Outdoor Film Coefficient

An outdoor film coefficient of 4.0 Btu per (hr) (sq ft) (deg F) has been used tentatively by those interested in the mathematical analysis of periodic heat flow through walls and roofs.[10] Exactly what is a proper value to use is open to question. Forced convection coefficients for surfaces that appear in THE GUIDE 1945 [8] (see Fig. 1, Chapter 4), were determined by experiments in which the test surface was installed near the end and in one side of a long duct. It has been shown [7] that this method does not entirely represent the conditions found in practice. Although the flow of air in ducts and the flow past surfaces such as walls are governed by the same laws, in the former case the flow is influenced by the enclosing surfaces of the duct. In the latter case this influence is lacking.

In order to more fully understand the importance of the outdoor film coefficient on the overall transmission coefficient of glass, new research is now under way at the A.S.H.V.E. Research Laboratory at Cleveland. In this project the relationship between the film coefficient and the length of a single smooth surface, measured in the direction of air flow parallel to it, is being studied. In addition, the effects of depth of window recess and of the projection and spacing of muntin bars are also being investigated. This study will reveal only the share of forced convection in the total coefficient.

TABLE 6—VARIATION OF T, A, AND R WITH REFRACTIVE INDEX
KL = 0.419; ANGLE OF INCIDENCE = 75 DEGREES

WAVELENGTH MICRONS	REFRACTIVE INDEX	T	A	R
2.00	1.490	0.342	0.367	0.291
0.89	1.526	0.341	0.362	0.297
0.28	1.560	0.340	0.356	0.304

Few data are available as to the rates at which an earthbound surface exchanges low temperature radiation with the sky either by day or by night. Not until this effect has been studied can the outdoor film coefficient be properly evaluated. Evaluation of this effect is of fundamental importance to the whole field of estimating heat gains or losses of buildings. An indication of the importance of this point in any attempts at analyzing building heat flow is given by some test results of A. F. Dufton.[11] Large scale guarded hot box tests in which a single glazed glass roof was exposed to the weather revealed that the rate of heat loss in Btu per (hr) (sq ft) (deg F) in winter during the period between 5:30 p.m. and 9:30 a.m. ranged between 0.99 and 1.46 and averaged 1.16, while during the 24 hour period the coefficient ranged between 0.78 and 1.29, averaging 0.98.

Thermal Conductivity of Glass

In order to simplify the heat flow calculations given in the preceding sections, there was assumed to be no resistance offered by the glass itself and that, therefore, the temperature of the glass was uniform. The report on Heat Transmission Through Windows [7] summarized the results of a number of investigations of the thermal conductivity of glass. Values were as low as 2.0 and as high as 9.6 Btu per (hr) (sq ft) (deg F per inch). Since there was little evidence by which to judge the reliability of the data, a value of 6.0 will be used here. If this value obtains and if the inside and outside film coefficients be respectively 1.5 and 4.0, the temperature

differential through a quarter-inch thickness of glass is less than one degree under the usual summer conditions. For thinner sheets, of course, the differential is less.

Reflection of Solar Energy from Nearby Objects

In the analysis that has been made, no account was taken of the fact that solar radiation is reflected to a window from lawns, concrete pavements and other outdoor surfaces. Unquestionably considerable radiation can emanate from such sources but present data are insufficient to justify an attempt to evaluate this effect.

Index of Refraction

In preceding sections it was pointed out that the index of refraction is a function of incident wavelength; this variation is known as dispersion. The International

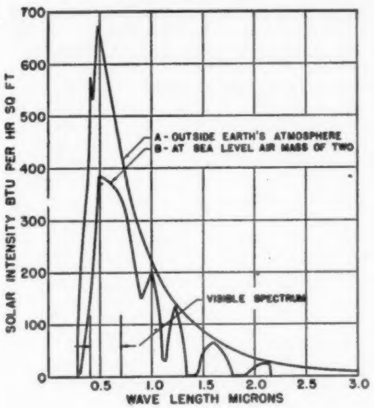

FIG. 12. STANDARD CURVES OF DIRECT SOLAR RADIATION INTENSITY ON A PLANE NORMAL TO THE SUN'S RAYS

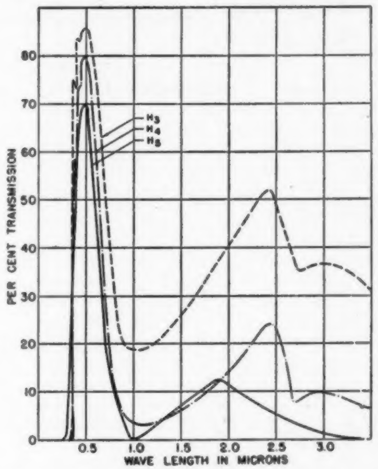

FIG. 13. SPECTRAL TRANSMISSION OF DIRECT NORMAL SOLAR RADIATION BY TYPICAL HEAT ABSORBING GLASSES

Critical Tables, Volume II, page 104, give the dispersion of glass for three wavelengths. The possibility that this variation would affect the absorption and transmission of glass was tested for solar energy incident at a 75 deg angle on glass having a KL value of 0.419. At this angle $g = 0.576$. Table 6 shows the results obtained.

It is seen that variations in refractive index have but a minor effect on transmission, less than 2 per cent deviation on absorption and that calculations based upon an index of 1.526 represent a good average.

Solar Energy Distribution and Its Relation to Transmission

Glass does not transmit all wavelengths equally well. This is particularly true of heat absorbing glass in the infra-red region. Since it is well known that the energy distribution of the solar spectrum varies with air mass (see Fig. 12 taken from the glass transmission report [7] and also Parry Moon's,[5] Proposed Standard Solar Radiation Curves for Engineering Use), the integrated transmission factor for one energy distribution may not be correct for a different energy distribution. In addition, the transmission of glass varies with wavelengths as shown by Fig. 13, also taken from

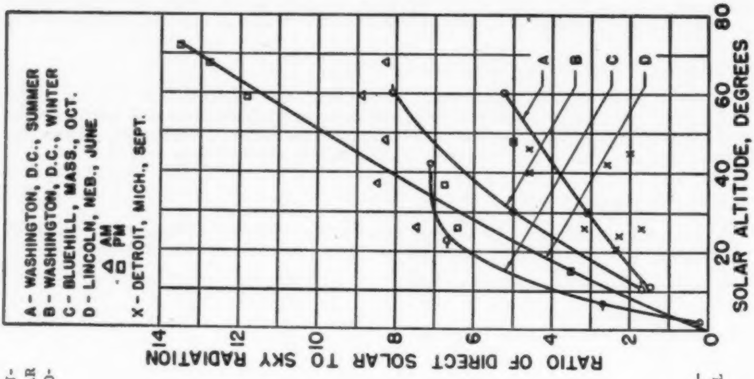

FIG. 14. (*Left*) RELATIONSHIP OF AIR MASS TO SOLAR ALTITUDE AND DIRECT SOLAR INTENSITY

FIG. 15. (*Right*) SKY RADIATION ON A HORIZONTAL PLANE

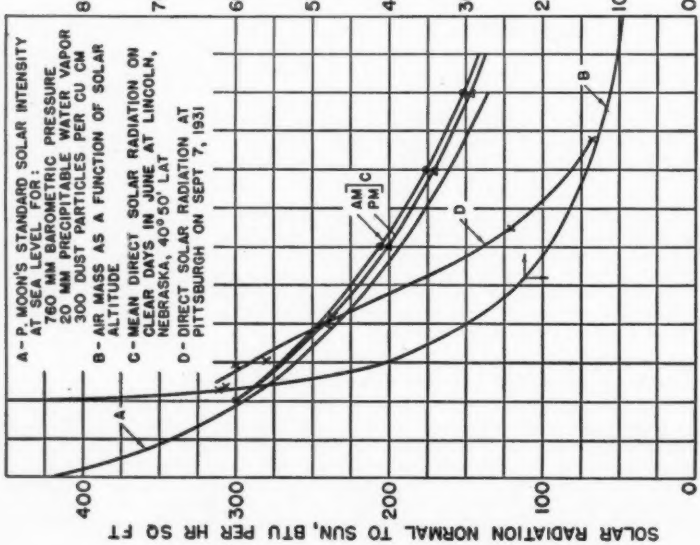

TABLE 7—VARIATION OF TRANSMISSION WITH SPECTRAL ENERGY DISTRIBUTION

AIR MASS	1	2	3
	TRANSMISSION		
Heat absorbing glass $KL = 0.584$	0.521	0.510	0.501
Plate glass $KL = 0.166$	0.786	0.783	0.782

reference 7. As it has been pointed out, the energy distribution is also related to the barometric pressure of a locality, the precipitable water vapor in the atmosphere and the dust concentration.

Spectral transmission curves were available for several heat absorbing glasses and several plate glasses. The transmission curves for an eighth-inch heat absorbing glass and a quarter-inch plate glass were selected and the average transmission calculated for air masses of 1, 2 and 3. The energy available for each wavelength range was multiplied by the transmission for that range. The summation of these values taken at 0.1 μ intervals from 0.4 through 2.0 μ and divided by the total energy gave the fraction transmitted. The results given in Table 7 show that the transmission is not significantly affected at normal incidence.

A third matter that requires consideration is the degree of polarization of the sky radiation. Sky radiation is solar radiation scattered by the air molecules, which are small compared to the wavelength. Since the ultra-violet radiation is shortest, the scattering of this radiation is greatest. The scattering effect has been found to be inversely proportional to the fourth power of the wavelength. Since the air molecules might well be considered as forming slits it is possible that the transverse vibrations in one direction would be suppressed to a greater extent than in the other. Such has been found to be true; the sky light is to some degree polarized. However, the difference is small enough that it can be neglected, particularly since the total energy is relatively small.

Solar Radiation Data

THE GUIDE, 1945,[8] at present incorporates tables of solar radiation on variously oriented surfaces. The values given in these tables (see Chapter 7, Tables 2, 3, 4 and 5) are the sums of the direct and the sky radiation. Because it was shown in the analysis presented in this paper that the direct radiation transmission depends upon the angle of incidence whereas the percentage of the total sky radiation transmitted does not, the two values should be considered separately. Further, in the report entitled, Heat Transmission Through Windows and Glass Panels,[7] it was pointed out that while the solar data given in THE GUIDE was representative of Pittsburgh, it could not be considered representative of other localities.

Moon [5] has worked out standard solar radiation curves in which the intensity is expressed as a function of air mass. Air mass simply refers to the thickness of the atmosphere and is very closely equal to the cosecant of the solar altitude. Obviously corrections are necessary for barometric pressure or altitude above sea level. These corrections are described in the report.[7] Fig. 14 reproduced here from the report shows Moon's recommended curve and data for Lincoln, Neb. Also it shows the observations made at Pittsburgh, Pa., on which in part the present Guide data are based. Included is a plot of solar altitude versus air mass. A summary was made of the means of observations made over a period of years at Lincoln, Neb.; Washington, D. C.; Madison, Wis.; and Boston, Mass. The observations for each of these stations for each air mass were averaged and in spite of differences in location and atmospheric conditions, the averages for the months of June, July and August lie within 5 to 10 Btu of Moon's curve. The averages for September are markedly higher by some 15 to 25 Btu. The curve, however, must be used with discretion. For example, Albuquerque, New Mexico, values are as much as 50 Btu higher at certain altitudes.

Unfortunately, our information concerning sky radiation is meager. Fig. 15, reproduced from the report,[7] gives some data on this point. Curves A, B and C represent

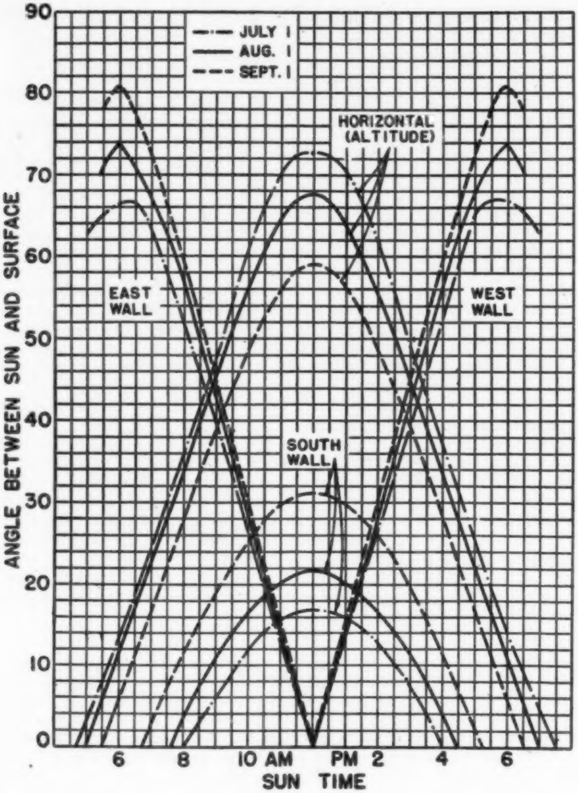

FIG. 16. CURVES GIVING THE ANGLE BETWEEN THE SUN AND A HORIZONTAL SURFACE AND WALLS FOR 40 DEG NORTH LATITUDE

the best data. This subject is now under investigation by the Solar Radiation Section of the U. S. Weather Bureau. In the absence of more complete data, curve A is recommended for summer cooling load calculations.

Solar Altitude

The angle of incidence of the sun's rays can be computed with the aid of well known formulae and astronomical observations. The altitude is a function of sun time, latitude, longitude and the declination of the sun. Of considerable help in these computations are the following bulletins, which may be obtained from the Government Printing Office, Washington, D. C. They are the American Nautical Almanac, which includes the Ephemeris of the Sun; and The Azimuths of the Sun, Hydrographic Office Bulletin No. 71.

Typical of the convenient curves that can be drawn are those of Fig. 16, reproduced from an A.S.H.V.E. Research Report.[15] The curves are for 40 deg north latitude

and show solar altitudes for three dates. The angle between the sun's rays and the surface is given instead of the angle of incidence. Therefore, when Fig. 16 is used with data given in this paper, the angle must be subtracted from 90 deg.

Heat Storage

The analysis of solar heat transmission through glass would not be complete without an investigation of the heat storage effects of glass, particularly of those types known as heat absorbing glass. Because both the outdoor temperature and the radiation intensity are continuously changing, how much time elapses between the change of these conditions and their influence on the cooling system? Although the question cannot be answered exactly, an example of the effect of an extremely sudden change can be cited.

Let it be assumed that a one-quarter inch heat absorbing glass is at 87.6 F, the steady state temperature for 90 F outside and 80 F inside. Then assume that the glass is suddenly exposed to solar radiation of 200 Btu per (hr) (sq ft) intensity, 56 per cent of which is absorbed by the glass. This is equivalent to raising the outdoor temperature to 118 F.

For thin sheets of uniform temperature the following equation gives the relationship between the time and the sheet temperature:

$$\theta = \frac{\rho VC}{A} \ln \frac{B - At_1}{B - At_2} \qquad (45)$$

where

θ = time, hours
ρ = specific weight, lbs per cu ft
C = specific heat, Btu per (lb) (deg F)
V = volume of sheet, cu ft
$A = h_o + h_{cri}$
$B = h_o t_o + h_{cri} t_i$
h_o = outdoor film coefficient (radiation and convection combined) Btu per (hr) (sq ft) (deg F)
h_{cri} = indoor film coefficient (radiation and convection combined) Btu per (hr) (sq ft) (deg F)
t_o = outdoor temperature, deg F
t_i = indoor temperature, deg F
t_1, t_2 = initial and final temperatures of sheet, deg F

In this example, it is assumed that $\rho = 148$, $C = 0.204$, $h_o = 4.0$, and $h_{cri} = 1.25$. As stated before, t_1 equals 87.6 F. The steady state temperature, 109 F, is reached in 34 min. This is, of course, an extreme case, one not likely to be encountered in actual installations. It is, however, important to consider in experimental work.

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DISCUSSION

CYRIL TASKER, Cleveland, Ohio (WRITTEN): Because at the time of preparation war conditions made it probable that this paper would not be presented before a regular Society meeting, a number of advance copies were sent to members of the committee and to others interested in this general subject, together with an invitation that they contribute written discussions to be published with the paper. The comments received are therefore included together with the author's closure.

W. E. ZIEBER, York, Pa. (WRITTEN): I think the form of this report is excellent. The method of presentation wherein you state all of the pertinent data for examples and then illustrate its use with examples will always be an excellent one. The appendices explaining the various methods of attack appeal to me very much. Reports of this type, even though this one is only a literature search correlating the information of such a search, will help immensely to prove to corporate managements that the A.S.H.V.E. is getting down to firm foundations in its research. The report is intelligible, leads to a logical conclusion and presents information although not proven by experimental investigation. This information is useful as a guide to what can be expected from such experimentation.

Some of the interesting facts presented, bring out some important phases of the effect of glass upon cooling load calculations.

For cooling load calculations in designing a system we need only the sum of the peak loads at the designated time which might be at night or in the daytime.

Since glass causes an almost instantaneous load, it is important that the direct radiation and sky radiation and the absorption ability of the glass be evaluated thoroughly

by experimentation, in order to be certain of their practical effects in cooling load calculations. We should also be sure that double windows are properly evaluated, both from the heat absorption and transmission standpoint.

If the sky radiation has a decided effect upon the north side of a building which has a lot of glass windows, this effect should be evaluated to see whether it must enter into the cooling load calculations.

If reflection of light enters this problem, it should be evaluated sufficiently to see what its heat load amounts to or what amount of heat load it prevents from entering the cooled space.

The present Guide data shows direct and sky radiation combined. The information contained in this bulletin would make it appear that these should be separated. I believe we should carefully weigh this point after we have evaluated sky radiation, because we should strive for simplification of calculations and not complications. I believe the numerical value of sky radiation is so small that it would be best to use for it an equivalent figure for direct radiation.

On the heat absorption by glass, I am sure we need to know how much heat can be absorbed by so called heat absorbing glass. It appears from the paper that the temperature of the glass will rise and in one of the cases mentioned it is shown that it could rise from a 90 F outside ambient, with 80 F inside, from an 89 F glass temperature to a glass temperature of 109 F. If this is a fact, it would appear that heat is transmitted from the glass to the 80 F room, also from the glass to the outside 90 F ambient. Is it possible that the only heat prevented from entering the room with a heat absorbing glass is the amount that is transmitted from the 109 F glass to the 90 F outside ambient temperature? If this were the case then it would be necessary to have some facts on the glass temperature rise on so called heat absorbing glass with certain sun intensities.

It was interesting to note from Fig. 14 the relationship between Pittsburgh and Lincoln data in regard to solar intensity. There is only about 10 per cent maximum difference between 25 deg and 60 deg solar altitude. From 25 deg solar altitude down this difference increases rapidly, until at 12 or 13 deg solar altitude Lincoln is double Pittsburgh. This shows the effect of smoke and moisture in the atmosphere is greater during the early part of the day, or the late part of the day, more so than when the sun is at its maximum altitude. This indicates the importance of having the proper solar radiation figures for the particular location. It may not have a great effect on glass calculations, but it will affect cooling load calculations considerably where the fly wheel effects of building materials and the cyclic unsteady state heat flow must be taken into account.

Since this analysis is made on the basis of unshaded windows, I believe we should know the effects of inside and outside shading, as well as the effects of different colors and different materials that might be used particularly in inside shades. This refers to any difference there might be between venetian blinds and materials used as draperies which might enclose a window to keep out the direct rays of the sun. It might be found that some types of inside shading of certain colors would be very highly effective in preventing the heat loads from the sun, or in reducing the total quantity of heat entering the cooled space. I believe this should be included in the study because the data now in THE GUIDE shows that there are appreciable effects from shading windows and it is our desire to provide information which will reduce the cooling load to a minimum in all cases, with the minimum of expense to the building owner.

I have always been under the impression that the window recesses had an effect only as they shade part of the glass from the sun and that all that should be counted is the amount of glass that is exposed to the sun's rays. It is possible that with heat absorbing glass convection currents will be established due to the heat of the glass and in that case the depth of the window recesses may have considerable effect upon the motion of these convection currents and their velocity and also upon the velocity of the wind on the outside of the building as it strikes the glass. The deeper the

recess the more resistant will it be to air motion tending to dissipate the heat. This could affect the external heat transfer coefficient of the glass and should undoubtedly be evaluated by some experimental work.

J. P. STEWART, Syracuse, N. Y. (WRITTEN): Mr. Parmelee has earned the gratitude of air conditioning engineers by his thorough analysis of transmission of solar radiation through glass. We have an immediate use for the data on heat absorbing glasses and on double glazed windows. The data for a single layer of ordinary glass agree very closely with the best data we have available. We look forward with great interest to the experimental work now being conducted on solar heat transmission through glasses.

The following comments are intended to provide suggestions for future experimental work: I agree most heartily with Mr. Parmelee that there is a need for measuring the absorptivity to solar radiation for various colors used on window shades. These colors should at least include white, cream, tan and green, used on roller shades; and aluminum, cream and light gray paint commonly used on venetian shades. At the same time it would be desirable to obtain the absorptivity of various colors of paints used on the exterior of frame houses, as well as several colors of stone and brick. Experimental work to determine the effectiveness of various window shades should include white, cream, tan and green roller shades and white, cream, aluminum and gray venetian shades. Tests should be conducted with the plane of the slats parallel to the plane of the glass, as well as slats set at an angle of 45 deg with the plane of the glass. When light colored slats of venetian shades are turned parallel to the glass, the shades should be more effective in reducing the solar heat gain to the room because more solar energy is reflected to the outside rather than being directed into the room. In practice, perhaps the majority of venetian shades are adjusted so that the slats make an angle of 45 deg with the glass. Experimental work with outside venetian shades should include cream and light colors. I should also like to see tests on outside venetian shades with the slats painted white on the side exposed to the sun, but with the opposite side of the slats (on the inside) painted a dark color, such as green. Theoretically, this two-color shade should be more effective in reducing the solar heat gain to the room than a shade painted white on both sides because less solar energy would be reflected from the inside surface of slats to the room.

I would like to see the Society analyze test data and recommend a design curve for solar radiation in summer which would be similar to Professor Parry Moon's curve A in Fig. 14. This curve probably would not be much different from that of Professor Moon's. Since this curve shows only direct solar radiation, sky radiation should be added in order to get the total solar radiation impinging on any surface. Sky radiation should be measured on vertical and horizontal surfaces for all the hours during the day on several summer days. The only practical way to establish a design curve is to do so by measurement of solar radiation and I would suggest that the Laboratory consider obtaining this information in cooperation with the U. S. Weather Bureau. Perhaps such a design curve could be established from data already in the Weather Bureau records and from data available in the A.S.H.V.E. Laboratory. The range of fluctuation of the solar constant rarely exceeds plus and minus 1 per cent.* The solar constant is the solar intensity incident on a normal surface located outside the earth's atmosphere and at a fixed distance from the sun, this distance being equal to the mean distance between the earth and sun. Because the earth is closest to the sun about January 1st and farthest about July 1st, the solar intensity just outside the earth's atmosphere on January 1st is approximately 445 Btu per square foot per hour, whereas it is only 415 Btu on July 1st. The mean value of the solar constant is 430 Btu. The significance of this is that a design value of solar radiation selected for use in the United States in July should be increased ap-

* Annals of the Astro-physical Observatory of the Smithsonian Institution, Vol. 6, p. 30.

proximately 7 per cent for use in the southern hemisphere in summer for January where the latitude is comparable to that of the United States. Direct solar radiation for the United States during clear weather in July is approximately 300 Btu on a surface normal to the sun at noon and at sea level; thus, about 28 per cent of the direct solar radiation is lost and scattered in passing through the earth's atmosphere. The principal cause for variation in solar intensity at the earth's surface is therefore due to atmospheric changes.

Examination of a curve published by the U. S. Weather Bureau ** shows that this depletion of direct solar radiation in the atmosphere is caused by:

1. Atmospheric scattering, approximately 16 per cent, by air molecules, water vapor and dust;

2. Atmospheric absorption, approximately 12 per cent, by oxygen, ozone, water vapor, carbon dioxide, dust, etc.

The foregoing 16 per cent and 12 per cent do not include an allowance for dust. The scattering and absorption caused by air molecules should be fairly constant for a particular elevation and solar altitude. If one considers a particular altitude and solar altitude, the principal factors which cause the solar radiation to vary at the earth's surface are water vapor, dust, ozone and carbon dioxide. For lower solar altitudes the amount of solar radiation scattered and absorbed in the atmosphere may be much greater than the 16 per cent and 12 per cent previously mentioned. Sky radiation is the portion of the scattered radiation which is directed toward the earth's surface, the remainder being directed into the vast space outside the earth's atmosphere. Probably ozone and carbon dioxide have relatively small effect in absorption and scattering, but by far the greatest variables are water vapor and dust. The depletion due to water vapor alone may be 10 or 12 per cent. However, this may be much more than 12 per cent under certain conditions. For a particular water vapor pressure at the earth's surface the amount of precipitable moisture contained in the atmosphere may vary as much as 200 or 300 per cent.†

There is, however, a definite relationship between any particular water vapor pressure and the corresponding average amount of precipitable water and I believe we would be more interested in average values. The design solar radiation curve should not consist of the averages of all days during the summer months, but should consist *only of averages on certain days when the dewpoint is within 5 deg of the dewpoint corresponding to the design dry-bulb and wet-bulb temperatures.* Only relatively clear days should be included. The depletion due to dust may be even greater than that due to water vapor.

Solar radiation data should be analyzed over a period of several months, in June, July, August and September, and a basic curve established from data obtained by one or more stations located in the Middle West in a region relatively free from dust, elevation of station not to exceed 1000 ft. Also similar data should be obtained in several of our industrial cities, such as Pittsburgh, Chicago, St. Louis, to determine the range of a multiplying factor to apply to this basic curve which would apply to cities. This multiplying factor may be in the order of 0.90 or perhaps 0.80 in some instances, depending upon the amount of smoke present. This total solar radiation (direct plus sky) in Boston, Massachusetts was 12 per cent less than at Blue Hill (Blue Hill is only ten miles south of Boston and elevation is 300 ft higher than Boston). This comparison was made at noon on certain days when the sky was clear in both Boston and Blue Hill. Data should also be analyzed from a station near the Gulf, in Texas or Florida and from a station located near sea level in a semi-desert or desert country. Data should also be obtained from stations at approximate elevations of 5,000 and 3,000 ft. Albuquerque, New Mexico would be a very suitable station for the 5,000 ft elevation.

** Review of U. S. Weather Bureau Solar Radiation Investigation—Irving F. Hand, *Monthly Weather Review*, December, 1937.

† Montezuma Pyrheliometry—C. G. Abbott, *Monthly Weather Bureau Supplement No. 27*, 1926.

I recommend the use of another Hydrographic Office publication, No. 214, *Tables of Computed Altitude and Azimuth* which gives the solar altitude and azimuth angles direct for various hours without having to make calculations. This publication consists of five volumes which include latitudes from 0 to 49 deg inclusive, each volume consisting of 10 deg of latitude. This book may be purchased from the Superintendent of Documents, Government Printing Office, Washington, D. C., at a price of $2.25 per volume. It is necessary to use the *American Nautical Almanac* to obtain the declination of the sun for the day and month desired.

L. K. JONES, Pittsburgh, Pa. (WRITTEN): The author of this paper has dealt with an extremely complex set of physical phenomena by means of mathematical treatment condensed in the form of curves which are the common language of the architect and engineer. It is, however, hoped that even further simplification with a sufficiently close approximation can be derived for usage in actual practice, since the computation as shown in the example given over a cooling season would become extremely bulky.

The use of the term *direct solar radiation* as given in Figs. 1, 2, 3 and 4 might be stated more exactly. For instance, it is not clear to the reader why, in the example given the *direct normal solar radiation* (283 Btu) was multiplied by the cosine of the angle of incidence to obtain the *direct energy* in the fourth paragraph of the solution, which was again multiplied by a 73 per cent factor derived from Fig. 1. It would appear from the discussion in Appendix I that the value of *direct solar radiation* referred to in Fig. 1 was the intensity in the path of the ray and the reduction by the cosine of the angle of incidence before using this step should be unnecessary. However, if the cosine term were not used, the transmitted energy would increase by over 50 per cent which is probably incorrect.

The author states in this paper and in a former report, that the indoor film coefficient would be the controlling factor in normal heat flow and that a 20 per cent change in the outdoor film coefficient would cause only a 3 or 4 per cent effect on the overall coefficient (U). However, it should be borne in mind that to secure an increase in inside film coefficient of 20 per cent would call for a change in air movement at 1/3 mph of from 200 to 300 per cent, while to change the outside film coefficient, the same amount at, say, 15 mph would only call for a 28 per cent increase in velocity. While it is true that because the inside film coefficient in a building is initially smaller by a four or five to one ratio, it should also be pointed out that the chances of wide variation would in turn be much less likely than for the weather side of the windows. As a matter of fact, when variations in wind velocities such as experienced in practice from practically still air to 20 or even 30 mph are considered, the control might easily change from the inside to the outside of the window.

J. EARL FRAZIER, Washington, Pa. (WRITTEN): My criticism of this paper covering work done by the author during the past months can be confined in most part to mere explanatory terms.

I should like to see the weight of the glass sheets described as grams per square centimeter or ounces per square foot, or described as a definite limit of variation between the weight of the sheets.

At the last meeting of the Committee on Glass, which was held at Boston in January of this year, almost everyone in attendance was in accord that more experimental work must be done on low temperature radiation. This is partially brought out in the transcript submitted to me.

Generally speaking, the paper should be helpful to engineers concerned with heating and cooling and also with the moving of air.

H. B. VINCENT, Toledo, Ohio (WRITTEN): This paper contains many worthwhile ideas but is open to the following criticism.

The analysis is quite correct but the data on solar radiation are so meager that the question arises as to the accuracy of the results. However, if adequate information concerning the intensities of solar, ground and sky radiation become available, this method of analysis will be very valuable. The direct measurement of the intensities of

solar radiation is not extremely difficult and average values could readily be found for a number of localities. These, combined with the present paper would comprise valuable engineering information.

H. C. Dickinson, Washington, D. C. (Written): This paper appears to contain a thorough analysis of the subject in regard to plain, vertical, unshaded windows. I have a feeling that the paper is a little too thorough for the purposes of engineering design. In general, there is too much stress on the need for accurate data, considering the fact that meteorological conditions vary so widely and have a great effect on heat transmission through windows.

I have not checked all the equations in detail, but the general methods involved in their derivation are sound. Referring to Fig. 11, the author states that this is a straight line when plotted on semi-log paper against the arithmetical mean temperature. This is only approximately true.

The author takes the emissivity of the inside walls, floor and ceiling of a room as 0.85. It should be practically unity, since so far as radiation to or from a window is concerned, the whole interior is essentially a black body.

Author's Closure: The writer wishes to express his appreciation to Messrs. L. K. Jones, Pittsburgh, E. W. Conover, Detroit and Dr. H. C. Dickinson, Washington, who called the writer's attention to several minor errors in the original draft which were corrected. He also appreciates the suggestion in a letter from J. N. Livermore, Detroit, that it should be emphasized that although the order in which two unlike sheets of glass are placed makes no difference in the percentage transmission or absorption, but the order does make a difference in the rate of heat transfer from the inner glass. This is explained by the fact that while a given combination absorbs the same percentage regardless of order, the percentage absorbed by the inner sheet is affected by the percentage absorbed by the outer. Equations 27, 28 and 29 show this.

Mr. Jones has brought out the point that there may be some misinterpretation of the term *per cent transmission* as used in Figs. 1, 3 and 9. Since this is expressed as a function of KL with angle of incidence as a parameter, the reader might assume that the term also allows for the reduction of the intensity of radiation normal to the sun's rays by the cosine of the incident angle. Such is not the case. Either the direct radiant energy per square foot of glass area is multiplied by the appropriate transmission percentage, or the direct radiation per square foot area normal to the sun's rays is multiplied by the cosine of the incident angle and the appropriate transmission percenttage to obtain the rate of radiant energy transmission per square foot of glass area. The same reasoning applies to the term *per cent absorption*.

Although the reduction of the per cent transmission by the cosine of the incident angle would simplify the computation, particularly if data similar to Fig. 14 were used, the writer has drawn curves of the total rate of heat flow through a given type of glass, including the effects of sky and direct radiation and temperature difference. These rates of heat flow are expressed as functions of the direct radiation and the parameters, angle of incidence and temperature difference. When more definite information regarding the variations in the transmissivity of commercial glasses has been obtained, these curves can be made available.

Similar curves for shaded windows will undoubtedly require a third parameter, azimuth, to properly express these transmission characteristics. The percentage reduction by shades is by no means constant, as reference to Research Report No. 1180 †† will indicate. Considerable fundamental data on the absorptivity of types and colors of surfaces are required, as pointed out by several contributors to the discussion.

The writer agrees with Messrs. Stewart and Zieber that one of the most important problems facing the Society is the establishment of proper radiation curves for various localities. Particular attention should be given to sky radiation, particularly if the data are to be used in computing heat gains in the so called solar houses. The analysis

†† Heat Gain Through Western Windows With and Without Shading, by F. C. Houghten and David Shore. (A.S.H.V.E. Transactions, Vol. 47, 1941, p. 251.)

given in this paper, however, is not, as Mr. Vincent suggests, dependent upon accuracy of solar radiation data. Spectral distribution of energy, as pointed out in Appendix III, has a negligible effect on the results.

Mr. Jones' comments with regard to the share of the outdoor and indoor film coefficients are well taken. The writer's statement was based upon the design wind velocity of 15 mph and the recommended outdoor coefficient of 6.0 Btu per (hr) (sq ft) (deg F). Low wind velocities would alter considerably the heat transfer rates from the surface of the glass to the indoors, if considerable solar heat were absorbed. This can be illustrated by Equations 40 and 41.

Dr. Dickinson has stated that the curve given in Fig. 11 is not exactly a straight line. This is true, but for all practical purposes it may be considered so and was drawn so. There is actually a very slight curve in the points and for that reason it should not be extrapolated. The maximum deviation from a straight line is about 1 per cent. With reference to the emissivity of the interior of a room, the emissivity factor of a room and window combination is given by the expression $Fe = \dfrac{1}{1/e_1 + A_1/A_2\ (1/e_2-1)}$ *where* e_1 and e_2 are the emissive powers of the glass and the room surfaces respectively and A_1 and A_2 are their respective areas. If glass has an emissive power of 0.94, this value is also the limiting Fe factor for large rooms. For small rooms Fe may approach 0.80. The angle factor, Fa, is unity if the window is in one wall only, since it is then totally enclosed. If there are windows in more than one wall, this factor becomes less than unity. Possibly a value of 0.90 for $FeFa$ would have been a better compromise than 0.80. Obviously, if the space were totally enclosed by glass, all of which was at the same temperature, the heat transfer from the glass by radiation would be zero. A number of tests of the overall transmission coefficients of single windows have been made in guarded hot boxes, the surfaces of which were covered with reflective aluminum foil. The results were roughly 50 per cent lower than those obtained by tests where *black* surfaces were used.

In Memoriam 1945

Name	Joined
Fred C. Black (*Life Member*) Chicago, Ill.	1919
Thomas D. Boyd, Cincinnati, Ohio	1937
Davis S. Boyden (*Life Member* and *President* 1937) Boston, Mass.	1909
Edwin A. Davies, Des Moines, Ia.	1944
William A. Evans, Cleveland, Ohio	1918
Amos Hard, Cincinnati, Ohio	1938
Charles G. Heydon, Detroit, Mich.	1927
Comdr. Ferry C. Houghten, Washington, D. C.	1921
K. C. S. Jacobsen, Philadelphia, Pa.	1939
John H. Kitchen (*Life Member*) Kansas City, Mo.	1906
Roman Kolakoski, Arlington, Va.	1942
Herschell J. Larsh, Indianapolis, Ind.	1945
Harold B. Lauer, Los Angeles, Calif.	1935
Richard J. P. Leinberger (F2/c U. S. Navy) Milwaukee, Wis.	1942
William E. Leland (*Life Member*) Berkeley, Calif.	1915
Thornton Lewis (*President* 1929) Newtown, Pa.	1919
Edward C. Malady, Indianapolis, Ind.	1943
Paul McDermott, Plainfield, N. J.	1943
Frederick D. Mensing (*Treasurer* 1931-32) Philadelphia, Pa.	1920
Samuel G. Neiler (*Life Member*) Chicago, Ill.	1898
Frederic W. Phillips, Jr., Brooklyn, N. Y.	1921
Robert E. Poehner, Indianapolis, Ind.	1928
Robert E. Purcell, Detroit, Mich.	1916
Frederic W. Robinson, Newark, N. J.	1944
William A. Russell (*Second Vice-President*) Kansas City, Mo.	1921
George C. Schwarz, St. Lous, Mo.	1943
C. B. J. Snyder (*President* 1907) Babylon, L. I.	1895
Sidney B. Strouse, Atlantic City, N. J.	1921
Perry West (*Life Member* and *Treasurer* 1924-25) Lexington, Ky.	1910
P. O. Wierenga, Grand Rapids, Mich.	1931

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